10/26/07 CO

MEMORANDUM

INTERMOUNTAIN POWER SERVICE CORPORATION

TO: George W. Cross

FROM: Dennis K. Killian

DATE: October 18, 2007

SUBJECT: Bid Evaluation and Recommended Contractor for Compressed Air Audit and Leak Survey

Technical Services recommends awarding the contract for a service air audit and leak survey to Compressor-Pump & Service. They are the lowest price bidder in full compliance with the specifications. Please find attached, Purchasing's bid transmittal document requiring your signature.

Bidder	Survey	Audit	Expenses	Total
Centrifugal Equipment Serv Corp	$9,995	No Bid	Included	$9,995
Compressed Air Management	$11,600	No Bid	Included	$11,600
Compressed Air Consulting	$5,200	$14,800	Included	$20,000
Cascade Energy Engineering	$8,423	$19,749	Included	$28,161
Compressor-Pump & Service	$2,778	$22,500	$5,635	$30,913
Ingersoll-Rand	$10,000	$19,000	$4,000	$33,000

The two lowest bidders did not bid the entire work scope. Compressed Air Consulting did not supply any of the requested bid information and Cascade Energy Engineering only bid eight hours of work on-site which is not enough to perform the necessary tasks.

We believe that Compressor-Pump & Service is the lowest responsible bidder. Compressor-Pump & Service provides a more detailed report, supplies guidance in design of solutions for identified problems, and they are entirely vendor neutral.

The original requisition was for $25,000. Your signature below will authorize the increase up to the bid amount.

Questions regarding this recommendation may be directed to Bret Kent at extension 6447.

George W. Cross
George W. Cross
President and Chief Operations Officer

10/25/07
Date

BK/JKH:jmj
Attachments

IP12_001470

8/21/08 [illegible]

MEMORANDUM

INTERMOUNTAIN POWER SERVICE CORPORATION

TO: George W. Cross

Page 1 of 1

FROM: Jon P. Christensen

DATE: August 11, 2008

SUBJECT: Approval of Purchase of New Plant Air Dryers

Please indicate below your approval to purchase Plant Air Dryers at a cost exceeding the approved requisition amount of $270,000. The final recommended bid came in at $456,804. Sufficient funds were accrued from last years budget to cover the increased cost.

The original requisition amount was based on an estimate from the consultant that performed the air audit in February 2008. The complexity of the installation requirements (the space/piping restrictions require a custom configured dryer) drove the cost up. We anticipated this increase during the bidding period and proactively accrued enough funds to handle the increased amount.

The current air dryers are the biggest drain on the service air system and this project is still justified based on the savings from reducing the amount of service air used and reducing the number of compressors in-service (see attached).

Any questions regarding this request may be directed to Bret Kent at extension 6447.

George W. Cross

George W. Cross
President and Chief Operations Officer

8/21/08
Date

JPC
BK/JKH:jmj

Attachment

IP12_001471

MEMORANDUM

INTERMOUNTAIN POWER SERVICE CORPORATION

TO: IPSC Supervision

FROM: George W. Cross George W. Cross

DATE: October 23, 2008

SUBJECT: Replacement of Pnuematic Air Movers with Electric Fans in the Tool Program

Please educate your crews in the importance of avoiding the use of pneumatic air horns/movers. Additionally, your help is requested in locating/returning pneumatic air horns currently in use and ensuring that future use is restricted.

In an effort to provide improved reliability of the Plant Compressed Air System, Technical Services has been directed to replace pneumatic air horns with electric fans in the IPSC Tool Program. The driving force behind this decision is that the use of 10 air horns during the summer months, for bearing or general cooling, requires 1,240 cfm and costs $58,000 for 5,000 hours per year. Replacement electric fans will cost $2,500 per 5,000 hours operated. However, the true cost is 1,240 cfm is 50 percent of a plant air compressor, which would most likely require the operation of an additional compressor which costs $100,000 per 5,000 hours operated.

The decision has been made to leave the pneumatic air horns in the tool system so that they will be available for use in the coal yard (the electric fans purchased are not explosion proof). However, they have been flagged as "**ISSUE TO SUPERVISOR'S ONLY.**" Please stress the importance of only using them where an electric fan is not safe or logical.

The fans purchased range in size and configuration. They were purchased with the intent to be a direct replacement for our existing air horns. Carts, stands, casters, and tripods were also purchased to make them as easy to transport and use as possible. The attached pages detail the specifics of the fans.

Thank you for your cooperation in this matter.

BK/JKH:jmj

Attachments

Distribution List:

Jon Finlinson
Van Stewart
Boyd Cowley
Joe Duwel
Keith Mangrum
Scott Robison
Richard Schmit
Darwin Cheff
Steve Draper
Lloyd Leavitt
Ed Purcell
Mark Shipley
Wes Bloomfield
Ken Lebbon
Will Lovell
Kelly Cloward
Hugh Loukinas
Jeff Payne
Lynn Labrum
Jeff Schena
Bill Little
Sid Ogden
Craig Teeples
Mike Marshall
Mike Mooney
Sylvan Lovell
Lorie Cloward
Jerry Hintze
Brook Pace
Mike Nuttal
Ken Nielson
Dean Wood
Aaron Nissen
Blaine Ipson
Cindy Jones
Craig Young
John Fritzges
Lance Johnson
Howard Scott
Chuck Chamberlain
Bill Keel
Vance Lovell
Roger Stowell
Vance Bishop
Gary Goold

Coppus VANO 175VC
IPSC T/N 7574
2 Available in GSB Tool Room
3/4 HP, 120V - 1,500 cfm
Complete with transport cart.
Tripod available: T/N 7575

Coppus VANO 250VC
IPSC T/N 7573
5 Available in GSB Tool Room
1 HP, 120V - 3,000 cfm
Complete with transport cart.
Tripod available: T/N 7575

These fans are ideal for use in bearing cooling or ventilating a confined space. Their size and configuration are similar to pneumatic air movers and they are intended to be a direct replacement. Tripods are available to assist in aiming the fan directly at a bearing or opening and raising it to a height suitable for the application. Carts are included when the fan is checked out. The carts are similar to hand trucks and make moving them very easy.

Coppus Air Max-12
IPSC T/N 7578
2 Available in GSB Tool Room
3/4 HP, 120V - 2,200 cfm

Compact fans are light (44 lbs) and quiet (74 dBA). Ideal for personnel or equipment cooling.

Coppus TA16
IPSC T/N 7576
3 Available in GSB Tool Room
1-1/2 HP, 120V - 5,000 cfm

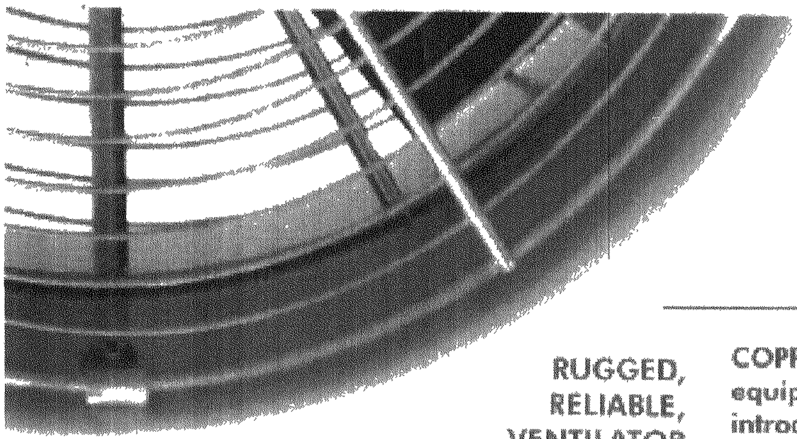
Large volume of air and long throw. Comes mounted on casters for easy movement. For ventilating large areas/enclosures.

Coppus Double Duty Heat Killer
IPSC T/N 7577
1 Available in GSB Tool Room
1 HP, 120V - 9,500 cfm
2 Speed Motor

Complete with stand and casters. Adjustable outlet guide vanes allow flow to be adjusted from a breeze to a jet. This feature makes it versatile for personnel cooling as the air velocity can be adjusted to suit the work being performed. The large volume of air moved by this fan and the stand also makes it ideal for ventilating large confined spaces.

Patterson PSF-22
IPSC T/N 7579
3 Available in GSB Tool Room
½ HP, 120V - 5,570 cfm w/100' throw

Complete with casters for easy moving. This is a ducted fan with a long throw. It has a misting pump for personnel cooling mounted on the fan stand. In low humidity areas the misting can drop the ambient temperature 20° F. This is a smaller version of what is being used for supplemental cooling on the Generator Step Up Transformer during the summer months.



COPPUS® VANO®
175CV, 250CV

RUGGED, RELIABLE, VENTILATOR MODELS

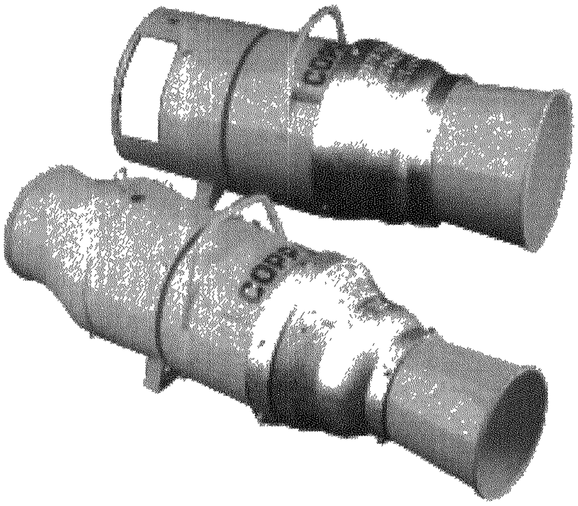
T/N 7574 MODEL 175CV: 3/4 HP
1,500 cfm (2,549 m^3/hr)

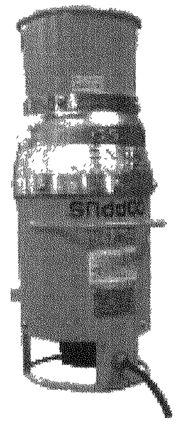
T/N 7573 MODEL 250CV: 1 HP
3,000 cfm (5,098 m^3/hr)

COPPUS ventilators revolutionized air moving equipment more than 60 years ago with the introduction of the VANO models. The VANO models offer a fixed guide vane design that delivers high volumes of air while maintaining static pressure for exhausting fumes and delivering fresh air. A rugged, durable, high-performance design makes the VANO models ideal for ventilating tanks, process vessels, tank cars, manholes and other confined spaces.

FEATURES

- Straightening fixed guide vanes for improved static pressure performance
- Accepts ducting at inlet and outlet ends
- Converts for exhausting fumes from bottom of tanks
- Available with totally enclosed (TE) or explosion-proof (EP) motors and compatible switch; all models supplied with 15-foot (4.572 m) power cord
- Heavy-gauge, powder-coated steel and cast aluminum construction
- Spark-resistant fan blades
- Automatic reset thermal overload protection standard on VANO 175CV and 250CV models
- Optional tripod and transport cart



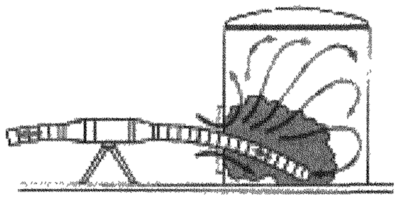


CONVERTIBLE DESIGN

By simply removing the inlet sleeves, the VANO 175CV and 250CV convert to vertical exhaust units

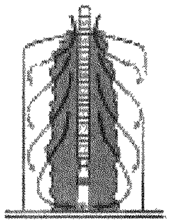
WITH INLET SLEEVE INSTALLED

The flexible duct can be attached to inlet and outlet ends. This allows fumes to be exhausted from a confined space, or fresh air delivered from a remote area.



WITH INLET SLEEVE REMOVED

Cut-outs on inlet end of CV models are exposed for exhausting heavier-than-air fumes from the bottoms of tanks, vats, drums and other confined spaces.



HAZARDOUS LOCATION MODELS

- VANO models are available with hazardous location (EP)* motors that meet NEC Class I, Division I, Group D and Class II, Division I, Groups E, F, G specifications

  *EP models do not include plugs

PERFORMANCE SPECIFICATIONS

AIR FLOW THROUGH FLEXIBLE DUCT - STRAIGHT RUNS [cfm (m^3/hr)]

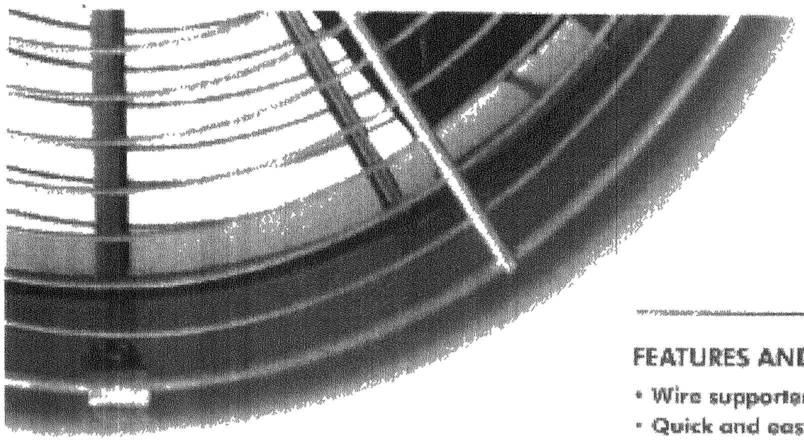
MODEL	10 foot (3.05 m)		20 foot (6.10 m)		30 foot (9.15 mm)		50 foot (15.25 m)	
	cfm	m^3/hr	cfm	m^3/hr	cfm	m^3/hr	cfm	m^3/hr
175CV	1,400	2,379	1,300	2,209	1,200	2,039	1,080	1,835
250CV	2,940	5,098	2,620	4,452	2,480	4,214	2,300	3,908

Performance schedule represents 60Hz synchronous speeds; 50Hz models perform at approximately 80 percent of listed schedules.

VARIOUS STATIC PRESSURES [cfm (m^3/hr)]

MODEL	FREE AIR		0.5 in WG 13 mm		0.75 in WG 19 mm		1.0 in WG 25 mm	
	cfm	m^3/hr	cfm	m^3/hr	cfm	m^3/hr	cfm	m^3/hr
175CV	1,500	2,549	1,350	2,294	1,280	2,175	1,230	2,093
250CV	3,000	5,098	2,620	4,453	2,480	4,214	2,300	3,908

ELECTRIC ITEM	dBA
VANO 175CV	93
VANO 250CV	92



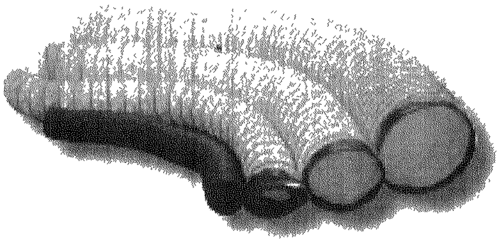
COPPUS® ACCESSORIES

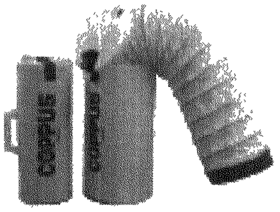
FEATURES AND SPECIFICATIONS (ALL VARIETIES)

- Wire supported, non-collapsible
- Quick and easy cinch belt securely fastens duct to blower housings and duct ends
- Integral rigid duct end allows easy coupling of duct without the need for separate splicer accessory
- Available diameters are 8-inch (203 mm), 12-inch (304.8 mm), 16-inch (406 mm), 20-inch (508 mm), and 24-inch (610 mm); larger diameters available on request
- Available lengths: 10-foot (3 m) and 25 foot (7.5 m); duct can be coupled together for longer runs
- Temperature range: -40 degrees F (-40 degrees C) to +250 degrees F (+121 degrees C)
- Meets UL-94 specifications for flame retardant material
- Retractable for easier, safer storage
- Source capture duct: close-pitched, wire-supported, features smooth interior walls for reduced flow restriction; available in 4-inch (102 mm), 5-inch (127 mm) and 6-inch (152 mm) diameters

FLEXIBLE AIR DUCT

A large selection of flexible air duct for a variety of ventilation applications is available. Our most popular heavy-duty duct features impregnated polyester material designed for harsh, industrial environments. Other options include economical light-duty duct, source capture duct and hazardous location, anti-static duct.

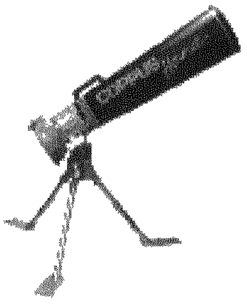




DUCT CANISTER

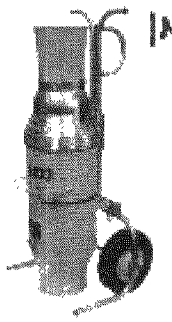
Extend the life of your duct with the protection of a COPPUS high-density, light-weight polyethylene canister; makes transporting and storage easier and safer.

Canisters for available duct sizes:
- 8" x 25' (203 mm x 7500 mm)
- 12" x 20' (305 mm x 6000 mm)
- 16" x 30' (406 mm x 9000 mm)



JECTAIR TRIPOD

For stationary, long-term use; rotates 360-degrees for precise direction of airflow and accommodates 3-HP and 6-HP Jectair sizes. Installs quickly and easily with two quick-release clamps. Large feet provide stability during operation, and spring-loaded legs fold up for easy transport and ~~storage.~~



INCLUDED W/ FANS

TRANSPORT CART

Heavy-duty cart allows easier transportation of VANO 175CV and 250CV ventilators (which can remain on cart during operation); includes crane-lifting loop.
WEIGHT: 25 lbs (11kg)

T/N 7575

VANO TRIPOD

Attaches to VANO 175CV or 250CV model; makes positioning of units and direction of airflow easier by rotating 360 degrees on a 45-degree plane; spring-loaded legs fold up for easy transport and storage.
WEIGHT: 19 lbs (9kg)

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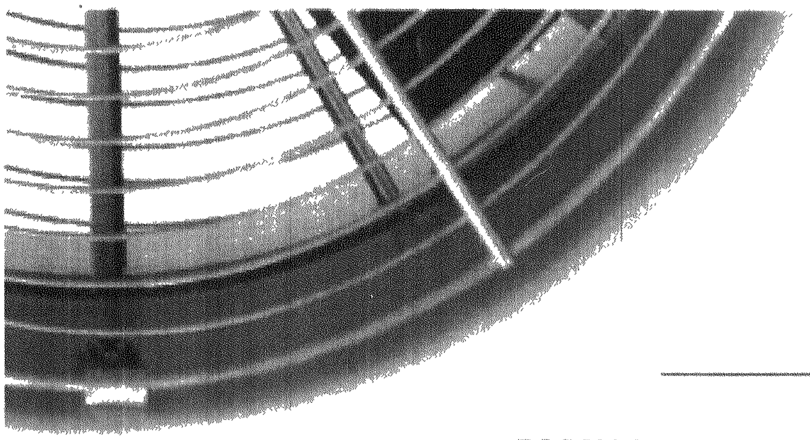
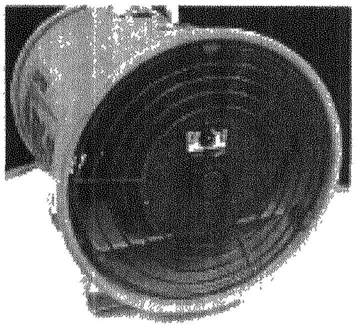


T/N 7578

COPPUS®
Air MAX-12

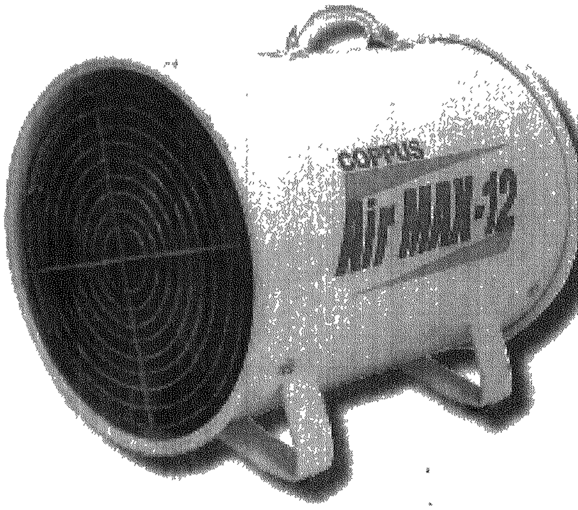


ECONOMICAL, HIGH-VOLUME TUBE AXIAL BLOWER

This 12-inch (305 mm), lightweight, rugged blower delivers up to 2,200 cfm (3,740 m³/hr) for confined space ventilation and fresh air supply.

FEATURES

- Rugged, all-steel housing construction
- Integral on/off motor switch
- Fixed guide vanes for improved performance
- Glass-reinforced, polypropylene, non-sparking fan blade
- 20-foot (6.1 m) cord with GFCI at plug end
- Anti-vibration foot pads



SPECIFICATIONS

MOTOR: TE 3/4 HP with integral on/off switch, 115V/6.8 amp, Class B insulation, auto reset thermal overload protection; attached 20-foot (6.1 m) heavy-duty cord with GFCI shut-off at plug end

FREE AIR: 2,200 cfm (3,740 m³/hr)

WEIGHT: 44 lbs (18.14 kg)

HOUSING: 18-gauge steel, powder-coated with carry handle and anti-vibration foot pads; rolled bead on ends for added strength and attaching flexible ducting; safety screens attached per OSHA guidelines

AIR FLOW THROUGH DUCT (STRAIGHT RUNS) cfm (m³/hr)

MODEL	FREE AIR		10 foot (3.05 m)		20 foot (6.10 m)		30 foot (9.15 m)	
	cfm	m³/hr	cfm	m³/hr	cfm	m³/hr	cfm	m³/hr
→ Air MAX	2,200	3,740	2,120	3.602	2,025	3,440	1,890	3,211

ELECTRIC	
ITEM	dBA
AirMax 12	74

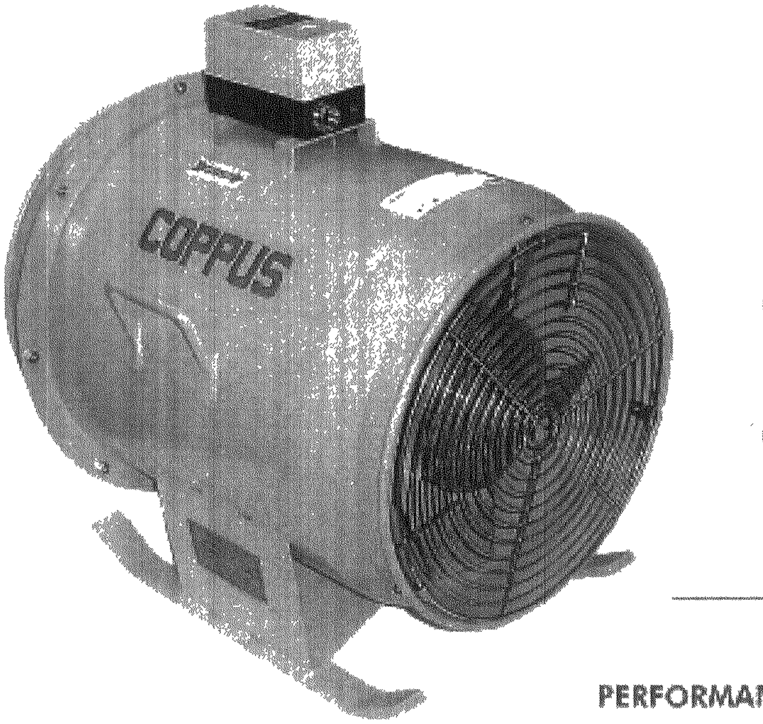
COPPUS® TA16 T/N 7576

DELIVERS LARGE VOLUME WITH HIGH-STATIC PRESSURE CAPABILITIES

The unique fan blade design not only allows exceptionally high air volume but also maximizes static pressure for better performance through longer runs of air duct. Typical ventilation applications include large tanks, tunnels, towers, and shipboard compartments; this fan also is ideal for product and process cooling.

FEATURES

- TE or EP motor
- Thermal overload protection
- Powder-coated, heavy-gauge steel housing
- Cast-aluminum or glass-filled fan blade provides spark-resistance
- Skid-mounted for stability (optional casters available)
- Duct can be connected at inlet and outlet ends



MODEL TA16-5500
2-HP, 5,500 cfm (9,345 m^3/hr) free air

MODEL TA16-5000
1 1/2-HP, 5,000 cfm (8,495 m^3/hr) free air

PERFORMANCE SPECIFICATIONS

AIR FLOW THROUGH FLEXIBLE-DUCT-STRAIGHT RUNS cfm (m^3/hr)

ELECTRIC	
ITEM	dBA
TA16	98

MODEL	10 foot (3.05 m)		50 foot (15.25 m)		100 foot (30.5 m)	
	cfm	m^3/hr	cfm	m^3/hr	cfm	m^3/hr
TA16-5500 2 HP	5,320	9,039	4,775	8,113	4,250	7,721
TA16-5000 1 1/2 HP	4,835	8,215	4,340	7,379	3,875	6,585

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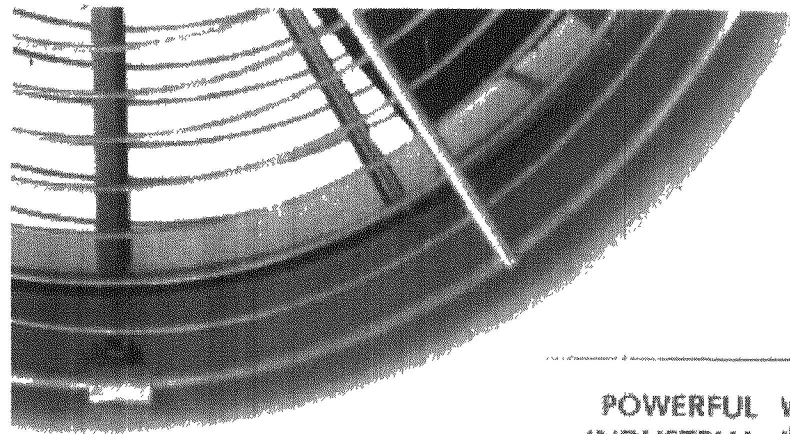
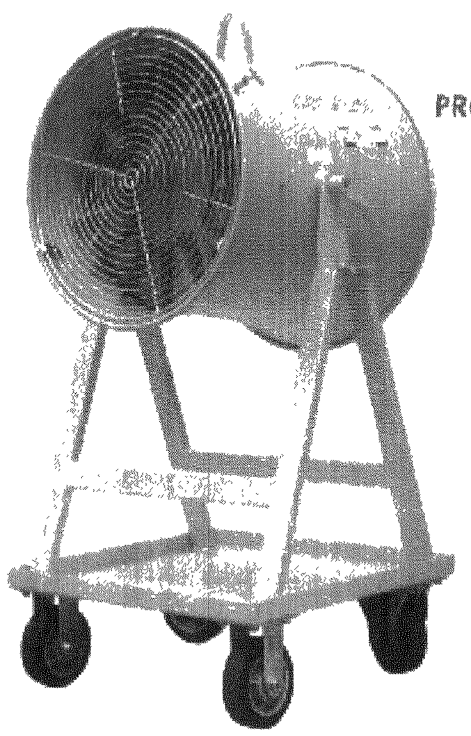


T/N 7577

COPPUS® DOUBLE-DUTY™ HEAT KILLER

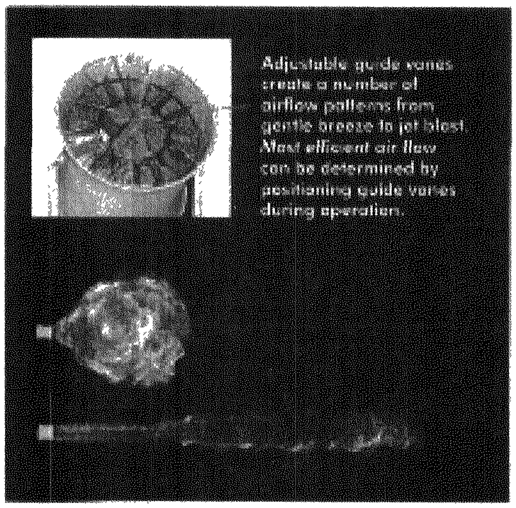
POWERFUL INDUSTRIAL FAN FOR COOLING PRODUCTS, PROCESSES AND PERSONNEL

With airflows up to 17,000 cfm (28,890 m³/hr) the Double-Duty™ Heat Killer (DDHK) is one of the most powerful and versatile portable air movers on the market today. The patented, adjustable guide vane design allows air flow control—from a gentle breeze for personnel cooling, to a concentrated jet blast for product and process cooling.



FEATURES

- Adjustable guide vanes allow varied air movement from a gentle breeze to a jet blast
- Available in 18-, 24- and 30-inch (457, 610, 762 mm) models
- 18-inch (457 mm) and 24-inch (610 mm) models available with two-speed, totally enclosed motors (115V only)
- Models available in floor stand or wall mount models
- Heavy-duty, rugged steel housing and frame
- Protective screens meet OSHA guidelines
- Available with TE and EP motors
- Hazardous location switches and motors meet NEC Class I, Division I, Group D and Class II, Division I, Groups F and G specifications
- Thermal overload protection on motors



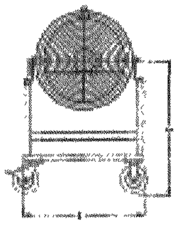
Adjustable guide vanes create a number of airflow patterns from gentle breeze to jet blast. *Most efficient air flow can be determined by positioning guide vanes during operation.*

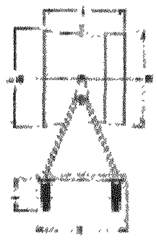
PERFORMANCE SPECIFICATIONS

MODEL	FAN SIZE		MOTOR		AIR VOLUME	
	in	mm	HP	rpm	cfm	m³/hr
18K03D	18	457	1/3	1,750	4,100	6,970
24K07D	24	610	3/4	1,750	7,100	12,060
24K10D	24	610	1	1,750	9,500	16,140
30K30D	30	762	3	1,750	17,000	28,890

FLOOR STAND DIMENSIONS - inches (mm)

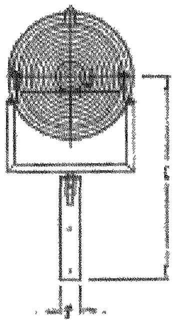
FAN SIZE	A	B	C	D	E	F	G	NET WT. lbs (kg)
18 in (457 mm)	23.0 (584)	21.3 (541)	36.5 (927)	24.0 (610)	24.3 (617)	7.5 (190)	21.3 (540)	140 (59)
24 in (610 mm)	29.6 (752)	28.3 (719)	39.5 (1003)	28.0 (711)	36.5 (927)	9.5 (241)	25.0 (635)	180 (82)
30 in (762 mm)	36.8 (935)	33.4 (848)	39.5 (1003)	28.0 (711)	36.5 (927)	9.5 (241)	28.0 (711)	230 (104)

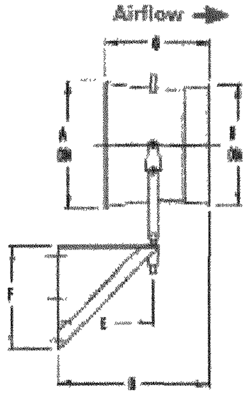




WALL MOUNT DIMENSIONS - inches (mm)

FAN SIZE	A	B	C	D	E	F	G
18 in (457 mm)	23.0 (584)	21.3 (541)	39.3 (998)	29.2 (742)	18.7 (475)	20.0 (508)	21.3 (540)
24 in (610 mm)	29.6 (752)	28.3 (719)	47.6 (1208)	36.4 (925)	22.9 (582)	24.0 (610)	25.0 (635)
30 in (762 mm)	36.8 (935)	33.4 (848)	52.7 (1338)	39.6 (1006)	25.1 (638)	26.1 (664)	28.0 (711)



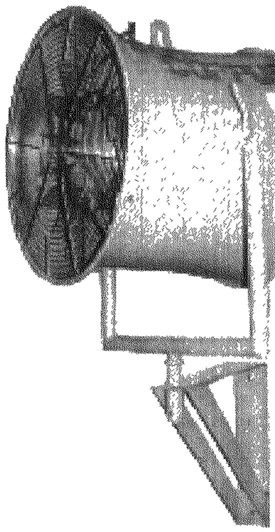


DDHK WALL MOUNT KIT

Easy retrofit wall mount kits are available for existing floor stand models; kits include wall bracket and fan U-bracket.

- Frees up valuable floor space
- Ensures permanent location
- Design permits 360-degree rotation with a total tilt of 155-degrees (90-degrees down, 65-degrees up)

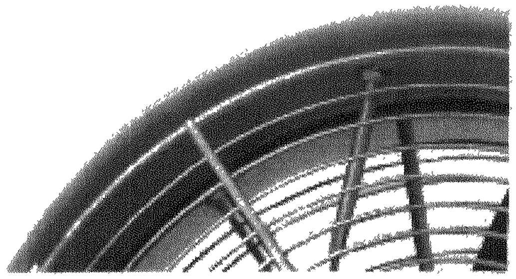
Electric	
ITEM	dBA
18K03D	72
24K07D	79
24K10D	85
30K30D	92



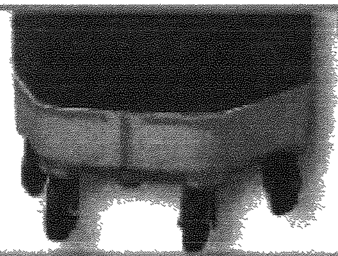
AIR VELOCITIES: fpm (m/min) at various distances from fan

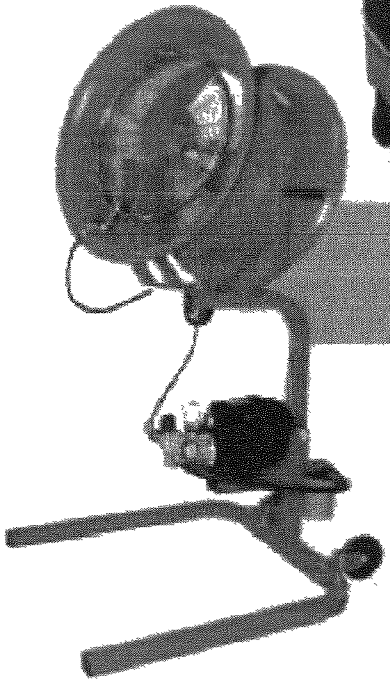
MODEL	10-ft (3 m)		30-ft (9 m)		50-ft (15 m)		70-ft (21 m)		90-ft (27 m)
	JB*	DF*	JB	DF	JB	DF	JB	DF	JB
18K03D	1100 (335)	425 (130)	450 (137)	260 (79)	350 (107)	180 (55)	275 (84)	135 (41)	--
24K10D	1675 (508)	670 (203)	840 (255)	375 (114)	560 (170)	280 (85)	350 (106)	230 (70)	185 (56)
30K30D	2250 (686)	1280 (390)	1000 (304)	520 (159)	900 (274)	340 (104)	700 (213)	310 (95)	475 (145)

*Jet Blast (JB) Diffused Flow (DF)



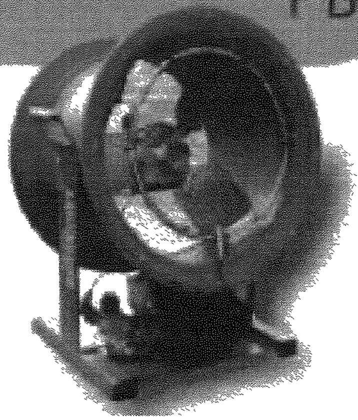
[illegible]	5000	1	31 1/2	[illegible]	8 x 8	[illegible]	[illegible]





Patterson's Portable Evaporative Cooler has a long air throw and can cool the ambient temperature up to 20 degrees Fahrenheit.

PBF/PSF*--Portable Fogging Fan



Model	CFM	HP	Width	Depth	I.D.	Shipping Weight	Distance Air Throw
PBF-14	2600	1/4	20"	16"	14 5/8"	78lbs/35.38	60'/18.28m
PBF-18	3800	1/2	24"	16"	19 1/8"	100lbs/45.35	70'/21.34m
PSF-18	3800	1/2	24"	16"	19 1/8"	126lbs/57.16kg	70'/21.34m
PSF-22	5570	1/2	29"	16"	22 1/2"	133lbs/60.33kg	100'/30m
PSF-26	7650	1	33"	18.5"	27"	157lbs/71kg	105'/32m
PSF-30	12,000	1	37"	22"	30 7/8"	180lbs/81.65kg	160'/48m

Cool ambient temperature significantly in the fan's air stream using fine mist.

IP12_001482

Intermountain Power Service Corporation
Compressed Air System Review: Executive Summary

Intermountain Power Service Corporation currently spends $775,071 annually on energy to operate the compressed air system at Delta, Utah. This figure will increase as electric rates are raised from their current average of 5.0 cents per kWh. The set of projects recommended below could reduce these energy costs by $385,214 or 50%. Estimated costs for completing the projects cannot be finalized until IPS obtains project cost quotes, but it is expected that the total cost will be yield a project with less than a 2-year payback or $770K.

PROJECT	SAVINGS PROFILE	ENERGY AND OTHER SAVINGS			TOTAL PROJECT COST ($)
		AVG kW	kWh	SAVINGS ($)	
AIR COMPRESSOR SUPPLY					
1. Reduce discharge pressure of compressor from 125 psig to 110 psig – increase flow 7% and increase turndown – 157 scfm each or 628 scfm, 4 compressors	Additional air flow using same amount of energy.				Part of Projects #2 , #6,and #7
	314 scfm (equivalent new capacity)	73.0	639,800	$31,990	
CAPACITY CONTROL					
2. Add central master control system with inlet guide vanes allow full turndown to meet conditions (15%) and lower electrical energy use 85%	208 kW	208	1,822,080	$91,104	TBD
AIR TREATMENT					
3. Replace four current heatless dryers with new modern similar-sized blower purge dryer with auto dewpoint demand controller	1348 scfm	257.4	2,254,680	$113,946 ($1,212)*	Est. Cost $220,000 + installation
4. Replace notched ball valve drains with level-activated drains (on 4 main water-cooled after-coolers)--2 drains per point of each compressor aftercooler separator (eight drains)	110 cfm	21.2	185,960	$9,298	$9,350 installed
DEMAND-SIDE SYSTEM					
5. Remove orifice plates on receiver entry to the casings on Units #1 and #2 Bag House and install appropriate regulators on the discharge line of each receiver.	After dryer 10 – 20 psig	This is not an energy issue but a reliability and production issues			

PROJECT	SAVINGS PROFILE	ENERGY AND OTHER SAVINGS			TOTAL PROJECT COST ($)
		AVG kW	kWh	SAVINGS ($)	
6. Eliminate excessive pressure loss in compressor area between compressor discharge and distribution system, reconfigure piping systems as required.	481 scfm (16 psig)	92.8	813,180	$40,659	TBD
	This is not only a direct energy issue but a project to enhance reliability and production				
7. Implement ongoing leak management program	855 scfm	165.0	1,445,460	$72,273	$20,600
Special Project 8. Replace air operated air horns used in warm weather with electric-operated units with same flow;	124 scfm per horn Equivalent of 10 horns 2190 hrs (summer season)	248.0 (2190 hrs)	543,120	$27,156 / yr	$20,000
TOTAL		1,065.4 kW	7,704,280 kWh	$385,214 per year	<$770K (<2-yr payback)

* Added electric cost from new dryer use of electricity directly.

Savings estimates depend, in part, on the capacity control system effectively translating lower air use into reduced electric cost. The current system does not have this type of unloading controls. With today's piping system, the controls will not accomplish this goal.

It also is important to note that other recoverable compressed air costs can also be considered, e.g., air system maintenance, water costs, and equipment life. Usually, the electric cost is between 50% and 75% of the total "variable compressed air costs." Associated maintenance and other costs are often more than 30% of the identified electric cost.

THESE PROJECTS AND THEIR PERFORMANCE ARE INTERCONNECTED. IF ALL THE PROJECTS ARE NOT IMPLEMENTED, THEN VERY LITTLE OF THE SPECIFIC PROJECTED SAVINGS WILL MATERIALIZE.

PROPOSED ACTION PLAN

- Reconfigure piping from dryers to main header eliminating 16 psig of pressure loss.
- Remove six orifice plates at feeds to Unit #1 and Unit #2 Bag House air receivers; replace with appropriate regulators on discharge of receivers.
- Replace heatless dryers with new similarly sized blower purge dryers.
- Install inlet guide vanes on all four compressors with compressed air central air management system.
- Replace notched ball valve drains on the aftercoolers separator drains with appropriate level activated electric or pneumatic actuated drains.
- Repair tagged leaks, implement a continuing leak repair program.
- Special: replace air operated cooling air horns use in the summer with electric operated, avoid 1240 scfm use / 5000 hours per year.

- After the system is reconfigured and stabilized, review the benefits in moving the air compressor inlet to receive cleaner dryer and cooler outside air.
    - This will have a positive impact on maintenance costs.
    - During the colder months up to another 150 scfm will be available assuming the motor can handle the 6% increase in power. This extra air may allow you to keep a third unit off line.
    - We do not recommend doing this unless the plant installs the proposed CEC control system with inlet guide vanes or something else equal.

PHASE 2 ACTIVITIES

After the basic system reconfiguration is implemented, a continuing review of the plant is recommended:

- Review regulator operations to ensure they are working and at lowest effective pressure.
- Review air-operated diaphragm pump with possible switch to electric-operated units if larger units or high cycles become the practice.
- Review the opportunity and payback of replacing older motors with high-efficiency units, whenever major electric motor repair is anticipated.
- Continue an aggressive leak tagging and repair program; quantify and <u>value</u> the leaks and report to management on a predetermined regular basis.
- After the system is reconfigured and stabilized, continue to review all dust collector installations.
- Continue to look for air-operated vibrator applications; monitor incoming equipment.

COMPRESSED AIR SYSTEM REVIEW

Prepared for

[redacted]

Intermountain Power Service Corp.

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February 2008

*Disclaimer: This report provides a general overview of the facility's compressed air system. As such, all data and analysis presented are estimates and should be only considered as guidelines. Final project specification and enumeration of potential savings and costs should be developed using appropriate compressed air system professionals. Cost and savings estimates and "totals" included in tables may reflect rounding.

TABLE OF CONTENTS

CHAPTER 1. COMPRESSED AIR SYSTEM REVIEW – OBJECTIVES

The REPORT SECTION of the Compressed Air System Review identifies specific projects to reduce air usage. These reductions usually translate into lower electric costs, improved system operation, and enhanced air quality and productivity. For a summary of results for this section, refer to the EXECUTIVE OVERVIEW at the front of this notebook.

For details of data gathered and work sheets completed, refer to the PLANT SURVEY SECTION of the notebook. For equipment performance and details, see the EQUIPMENT SECTION. For project cost estimates, refer to the PROJECT COST SECTION. For additional information and articles, see the MISCELLANEOUS SECTION.

The primary objective of the review is to provide a comprehensive list of specific measures needed to improve compressed air system operation and cost-effectiveness in the short- and long-term. The review addresses these topics:

- Review appropriateness of major equipment pieces in the compressed air system to produce the right quality and quantity of usable compressed air at an acceptable efficiency
- Develop a load profile of compressed air production
- Identify current electric power cost per cfm in order to establish a baseline for evaluating potential projects
- Evaluate characteristics and appropriateness of the use of a central compressed air capacity control system
- Outline plans for an ongoing leak management program
- Identify savings potential in use of air saving devices such as nozzles and auto drains
- Identify savings potential in replacement or re-evaluation of "potentially misapplied air" such as cabinet coolers, vacuum pumps, and bearing cooling
- Identify critical areas, if any, to utilize planned storage in the system:
    - Create effective storage, if required, for capacity controls
    - Establish stored volume to offset identified peak demand — local in system or off system
    - Establish stored volume to help set up proper use of pressure/flow controller
    - Create effective demand-side storage, as required, at critical points
- Estimate benefits of recommended savings measures, including reduced electric consumption and maintenance costs and improved productivity and system operation.

IP12_001488

CHAPTER 2. CURRENT AND PROPOSED SYSTEM REVIEW

2.1 CURRENT SYSTEM BACKGROUND

The power plant at Delta, UT, owned and operated by Intermountain Power Service Corporation, is a coal-fired plant with two basic power generating units. The compressed air supply is anchored by four 700-hp (660-hp) class, 3-stage centrifugal compressors with a rated flow of 2,242 scfm each at 100 psig. The compressed air then goes to four Pall Trinity (Pneumatic Products) heatless-type, twin tower desiccant dryers rated for a similar flow at 100 psig, 100°F inlet air.

The compressed air is used for instrument air throughout the complex and service air. The areas to be investigated are:

- The baghouse water and air flow – calculated cylinder air – leaks – air horn – evaluate piping – orifice restriction plates, etc.
- The appropriate use of compressed air dryer with regard to operating cost and required pressure dewpoint.
- Lime preparation compressed air facility.
- Water treatment compressed air supply.
- Boiler control air compressed air supply.
- Stack compressed air supply.
- Coal car unloading.
- Sludge transfer.
- Scrubber compressed air supply.
- Central air manifold.
- Crusher building.
- Coal yard transfer buildings.
- Limestone unloading.

Overall, the basic air system review is to:

- Document where reasonably feasible where the compressed air is being used and specific air where air conservation actions would be meaningful.
- Identify and tag compressed air leaks throughout the system.
- Analyze the dryer for the compressed air with regard to performance capability of supply reliable production and recommend any replacement or upgrades that may be in order.
- Evaluate the current FS/Elliott 3-stage centrifugals in a similar manner.
- Review the impact of compressed air inlet location – second floor from the area where the dryers are located.
- Review opportunities that may exist from unit performance upgrades and/or modern microprocessor control systems, etc.

IP12_001489

Setting the Baseline

The following actions were taken to establish the baseline for flow and pressure.

1. Temperature readings were taken on all units with an infrared surface pyrometer. These were observed and recorded to relate to the unit's performance, load conditions and integrity. The findings were recorded on the table of compressor supply operating data that follows.

2. Critical pressures including inlet and discharge were measured with Ashcroft digital calibrated vacuum and pressure test gauges with an extremely high degree of repeatability. Findings were also recorded in the table of appropriate compressor supply operating data specific pressures were taken and logged at points (see drawing). Plant personnel measured and logged unit amperage, voltage, and power factor simultaneously and operating kW was calculated from this data and compared to OEM data at the same discharge pressure.

3. Flows were measured after the compressed air dryers with heated wire-type, thermal mass flow meters and logged with MDL multi-line loggers. Results were compared to the OEM-supplied original test data and corrected for location conditions at the measured pressure operating times. Like most electric power plants, the Intermountain Delta plant runs 24 hours a day, seven days a week, or 365 days per year except for planned outages.

4. The normal full production includes running three to four compressors and four compressed air dryers at full load for 8,760 hours per year. The cost of compressed air to calculate and reduction paybacks is $50 per megawatt or $0.05 kWh.

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Phone: (740) 862-4112
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Ambient Pressure: 12.4 psia
Ambient Temp: 35°F
Ambient RH: 30%
Room Temp: 75°F
Date: 5 February 2008
Shift: First/Second Shifts

CENTRIFUGAL COMPRESSOR MEASURED OPERATING DATA

Compressor Unit	1D		1C		1B	1A	
Model	310DA3		310DA3		310DA3	310DA3	
Inlet Air Temp °F	95°		90°			87°	
Inlet psia (estimated)	12		12			12	
FL Flow (scfm)	2,242		2,242			2,242	
Capacity Control Type	IBV/BOV		IBV/BOV			IBV/BOV	
Discharge Pressure (PG) (psig)	119**		119**			119**	
Discharge Pressure (TG) (psig)	119		119			119	
Nominal Set Point (psig)	120		120			125	
Amp Limiters Set Points (min ___ / max. ___)	--	--	--	--		--	--
H_2O In °F / H_2O Out °F Unit	76.8	99.6	76.5	99.4	--	73	98.3
Air Temp 2[nd] Stage In/Out °F	121	295	136	255		130	280
Air Temp 3[rd] Stage In/Out °F	121	250	120	228		118	265
Calculated Full Load kW/amps	522.12 / 50.25		522.12 / 50.25			522.12 / 50.75	
Measured Full Load kW	--		--			--	
Percent Open IBVposition/BOV*/temp	Open	153°F	Closed	90°F		Closed	81°F
Percent Open IGV	--	--	--	--		--	--
Estimated Average Flow inc blowoff	2,242		2,242			2,242	
PKG Disch Air Temp °F	221°		258°			248°	

General Comments:
Two heatless dryers were off line – 674 scfm (purge not used) and dryers were in bypass (less pressure loss).
IBVs are all full open – IC is 10-15° off wide open but not moving.
*Amperage calculated at 6,600 volts/90 PF.

After-cooler							
H_2O In °F / H_2O Out °F	73	82	73	86	OFF	248	83
Air In °F / Air Out °F	221	84.2	258	83		77.4	81

IP12_001491

Figure 1. IPSC Control Air / Non-Essential

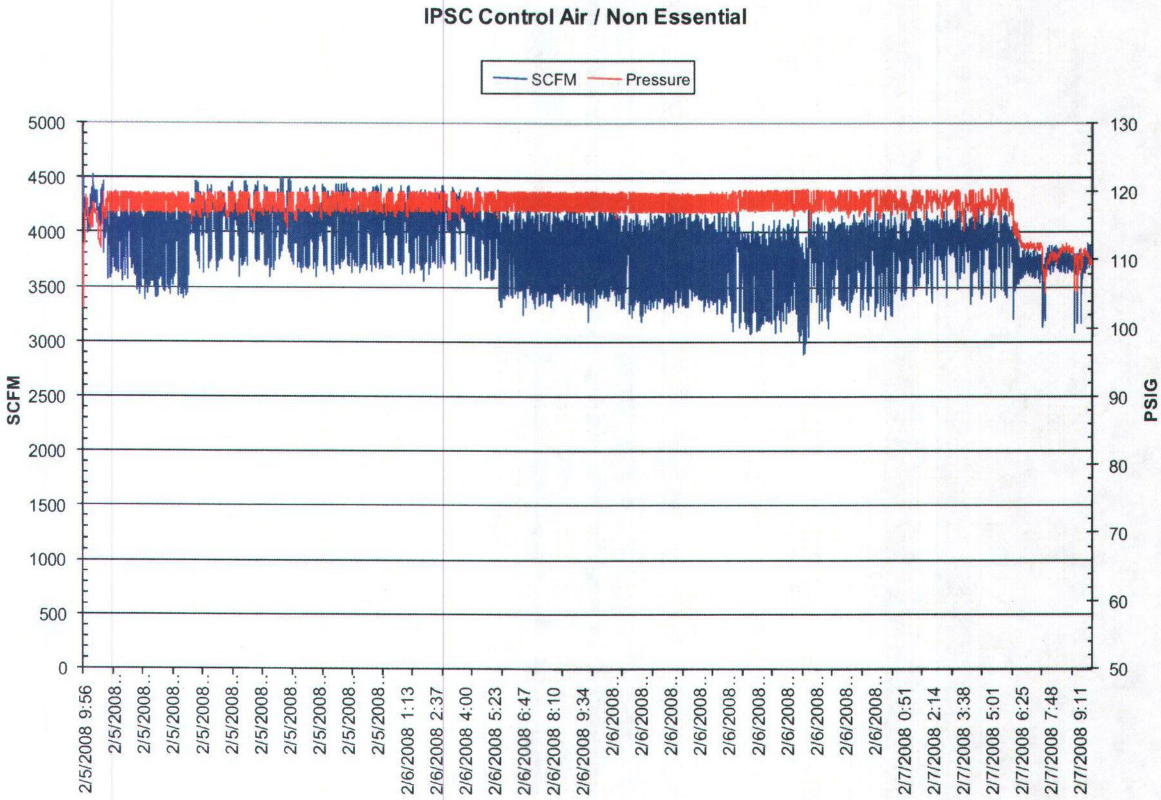
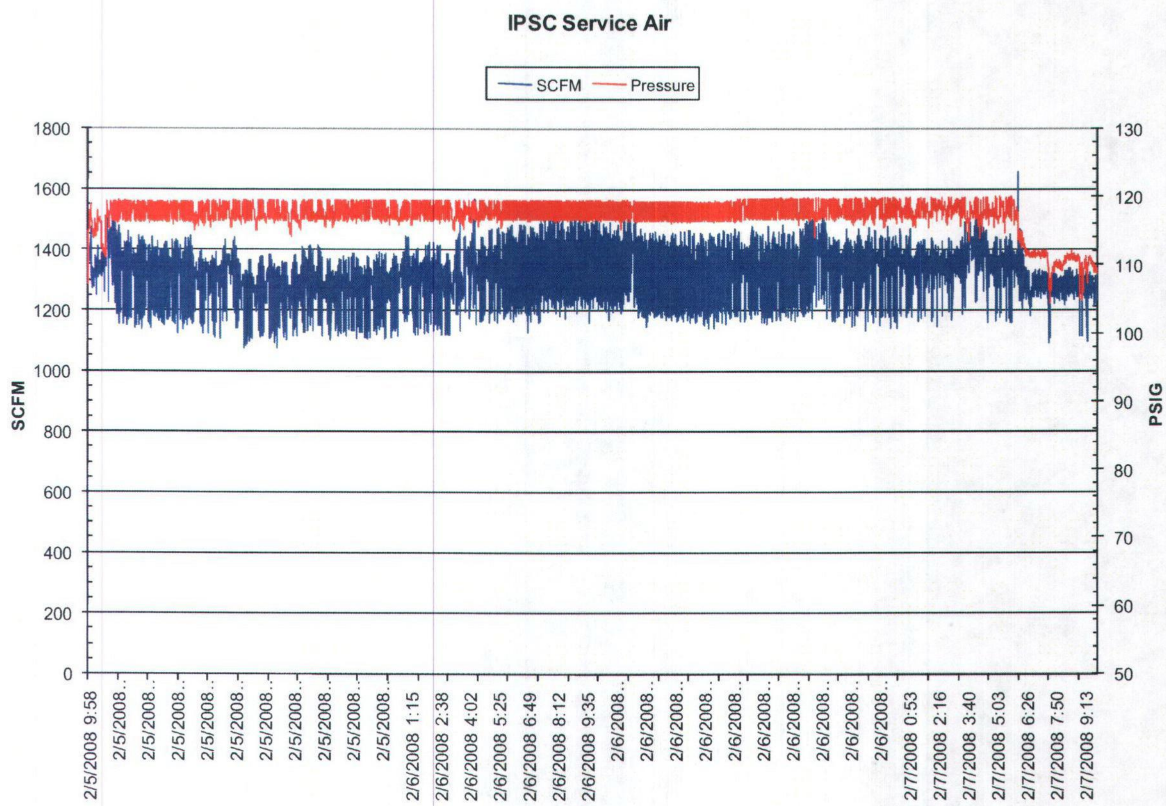


Figure 2. IPSC Service Air



IP12_001492

Figure 3. Coal Yard Flow
Isolated from 2008 to 2025

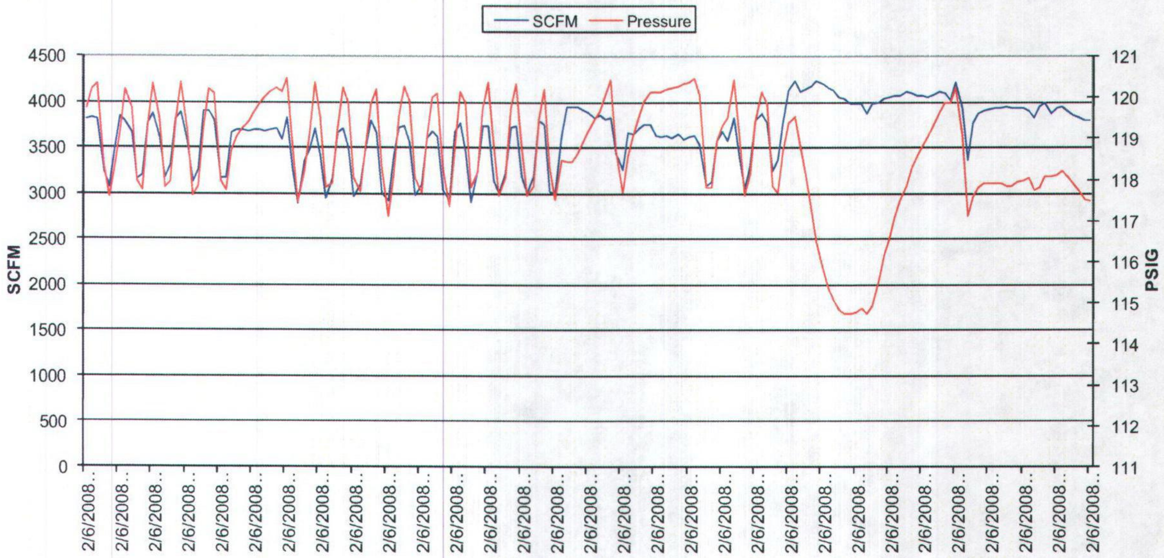


Figure 4. Water House Flow 2
Isolated Slaker Air Ejectors
Isolated from 2055 to 2122

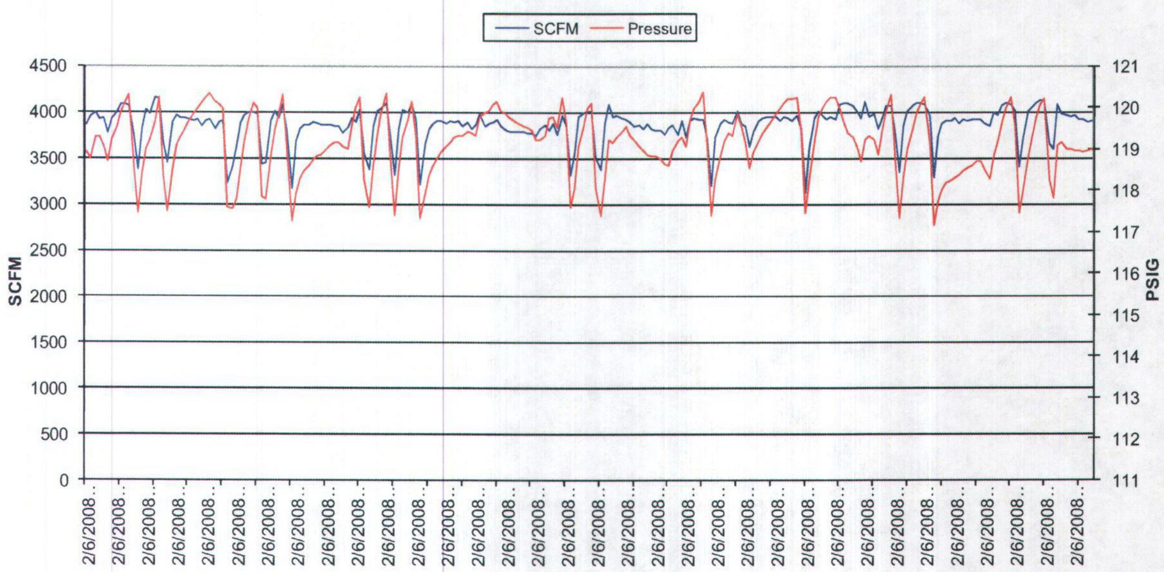


Figure 5. Water House Flow 1
Isolated from 2000 to 2018

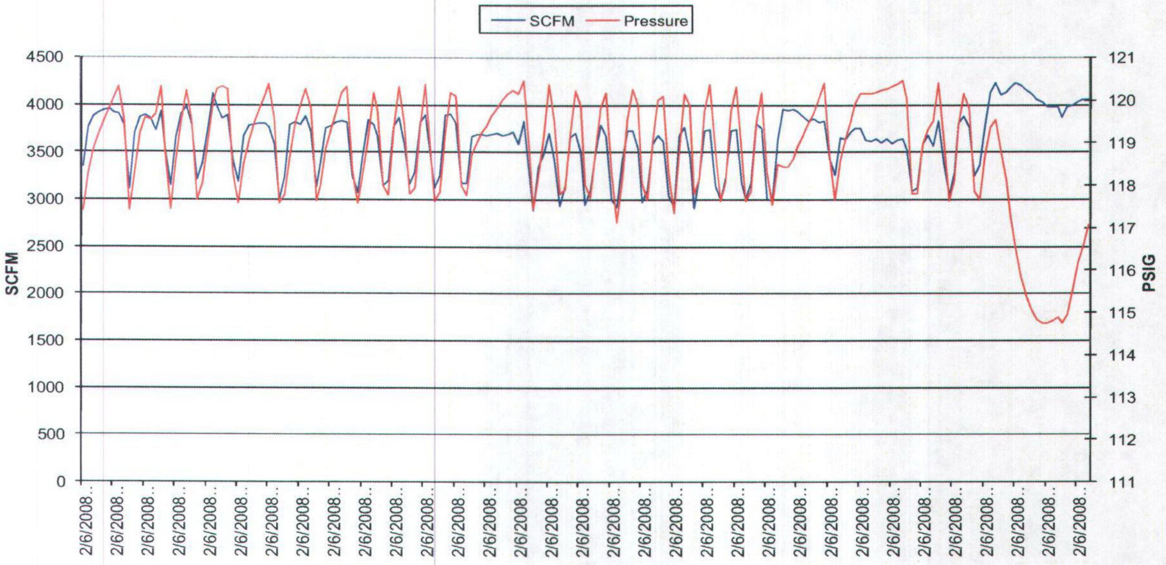
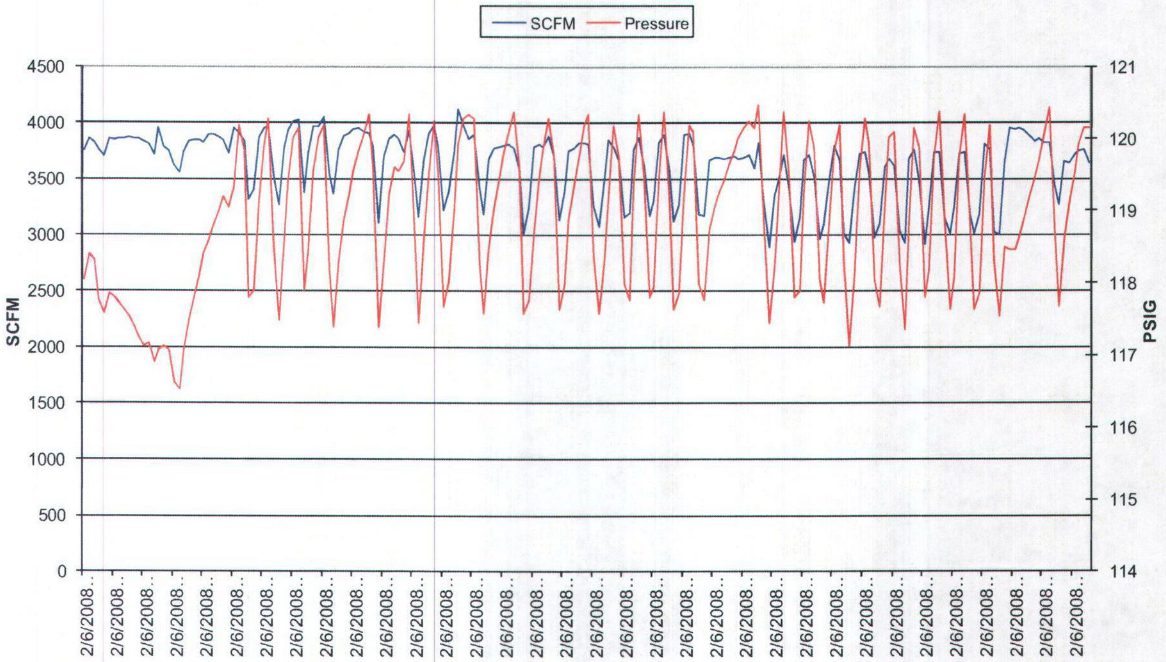


Figure 6. Sludge Flow
Isolated from 1955 to 2012



IP12_001494

The preceding chart shows the operational data of the three 700-hp class Elliott centrifugal units had while running from 9:00 a.m. to 10:00 a.m. This data reflect the operation with two dryers off, and therefore, 674 scfm of purge air demand, which is normally on the system – was not!

There are no observable indicators for the BOV action, but measuring temperature of the blow off manifold is an excellent indicator. As the high temperature discharge air enters the blow off, it raises the temperature of the line and manifold.

Reviewing the highlighted table, you will see that Units #1A and #1C have manifold temperatures consistent (82°F / 90°F) with the pipe temperature and the ambient indicating the presence of little or no hot bypass air. On the other hand, Compressor #1D has a significantly higher manifold temperature (153°F), indicating a continuing flow of hot bypass air.

Running the Bag House flow test to determine the affect of the air horns:

• Compressor 1D	IBV	Full open	Blow off manifold temp = 153°F
	BOV	Open	1,342 scfm
• Compressor 1C	IBV	Full open	Blow off manifold temp = 90°F
	BOV	Closed	
• Compressor 1B	OFF		
• Compressor 1A	IBV	Full open	Blow off manifold temp = 82°F
	BOV	Closed	

Referring to the following drawing of the basic piping system serving the Bag Houses:

- The air supply line to the non-essential air was shut off. All metered flow air was going to the bag house. Average non-essential air flow is 1,165 scfm.
- The bag houses were run with the air horns on. Each horn runs 10 seconds approximately every 4 ½ minutes. The air horns are computer controlled. There were three sets of air horns operating in the two units. Average air demand is 2,863 scfm with air horns on.
- The bag houses were then run without the air horns operating. The flow volume fell to an insignificant level.
- The service air remained a very steady 1,365 scfm.

During the Bag House test (Units #1 and #2), the compressors operation was observed:

- Total air flow on line was 6,735 scfm.
- 9:20 a.m. to 9:30 a.m. – system ran supplying all air to the Bag House, non-essential, and service air:
- Total air flow: FM1 (4,028 scfm) and FM2 (1,364 scfm) = 5,393 scfm.
- 9:31 a.m., we shut off the valve feeding the non-essential demand (see line drawing). FM1 fell to 2,863 while FM2 remained the same.

Conclusions (see charts)

- Non-essential air at this time had a value of 1,165 scfm (4,028 – 2,863).
- The 1,165 scfm reduction did not unload compressors 1A or 1C. It did increase the blow off in Compressor 1D to almost 100% flow.
- Blow off manifold temperature was 240°F.

Other Conditions

Two of the four desiccant dryers were off and bypassed. This had the following effect:

- With four dryers running, another 674 scfm will be required from the compressor to supply the purge air for the two dryers.
- The pressure loss, which with the bypass is negligible, will become more significant.
- Utilized properly, the dryers can begin to deliver dry air to the system to evaporate the condensate loaded in while they were out of service.

Figure 7. IPSC Delta, UT

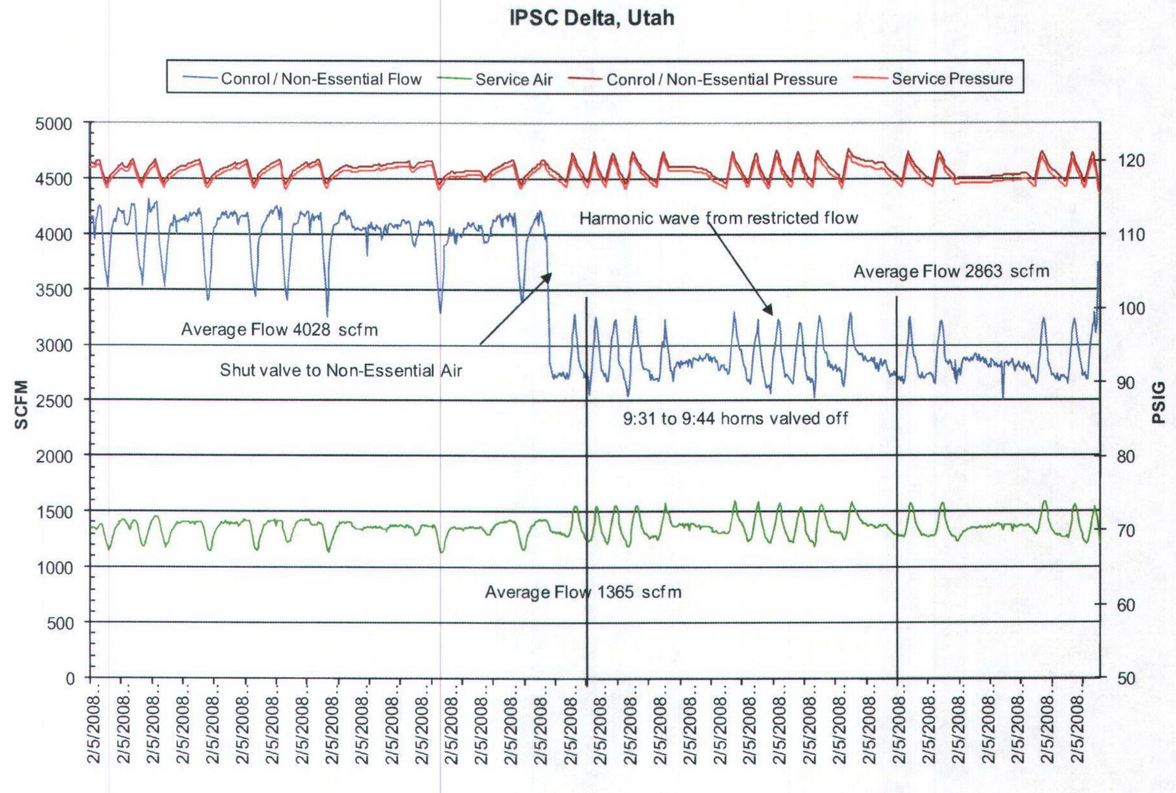


Figure 8. Bag House Test: Metering 9:10 a.m. to 9:43 a.m. 02/05/2008

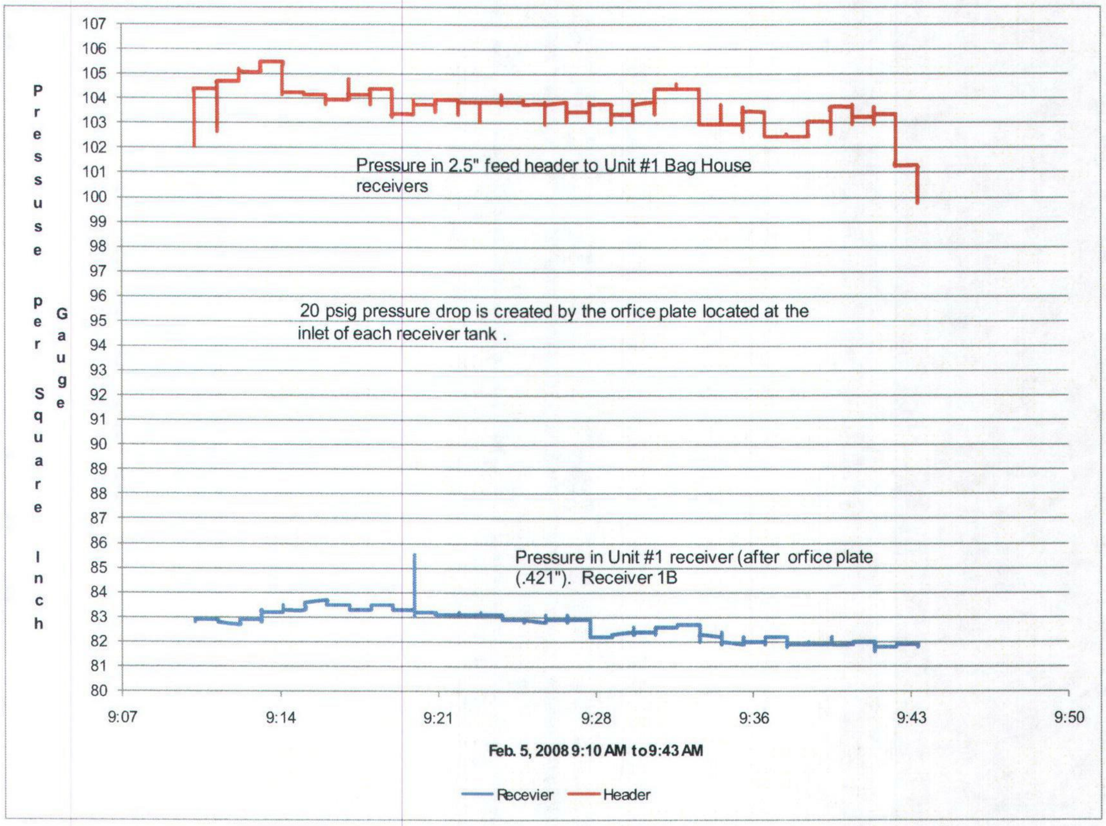
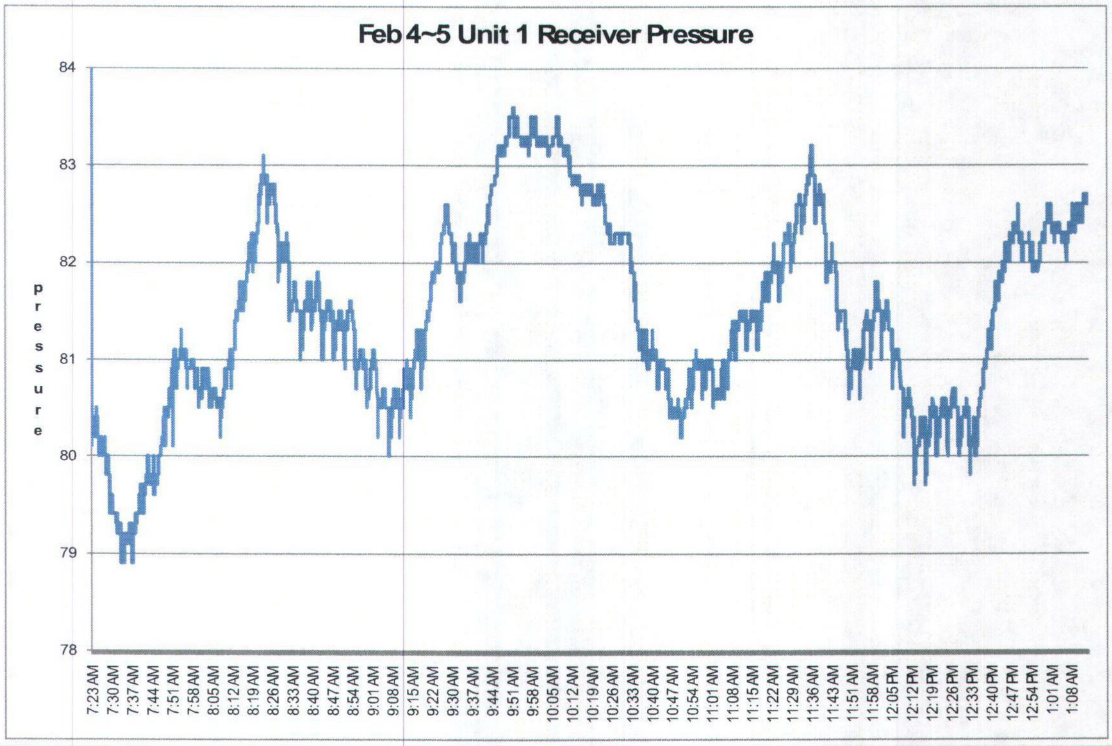


Figure 9. Unit #1 Receiver Pressure



IP12_001497

Figure 10. Unit #1 Header Pressure

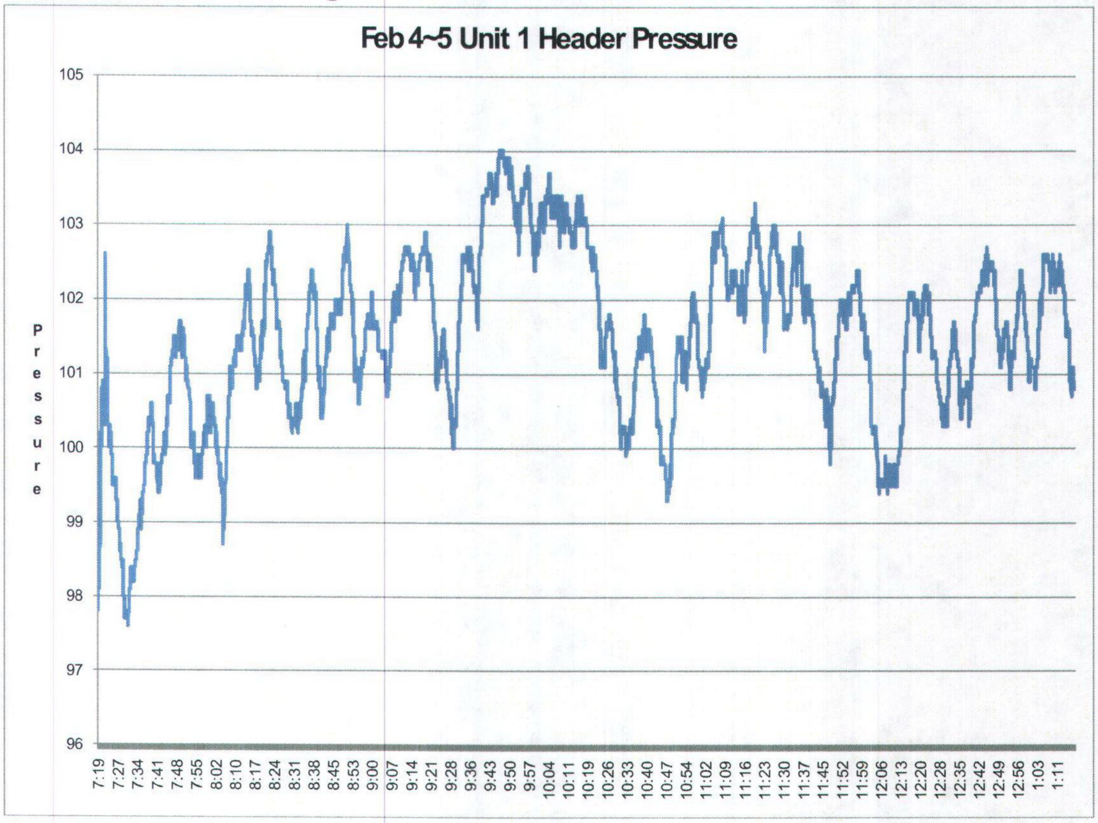
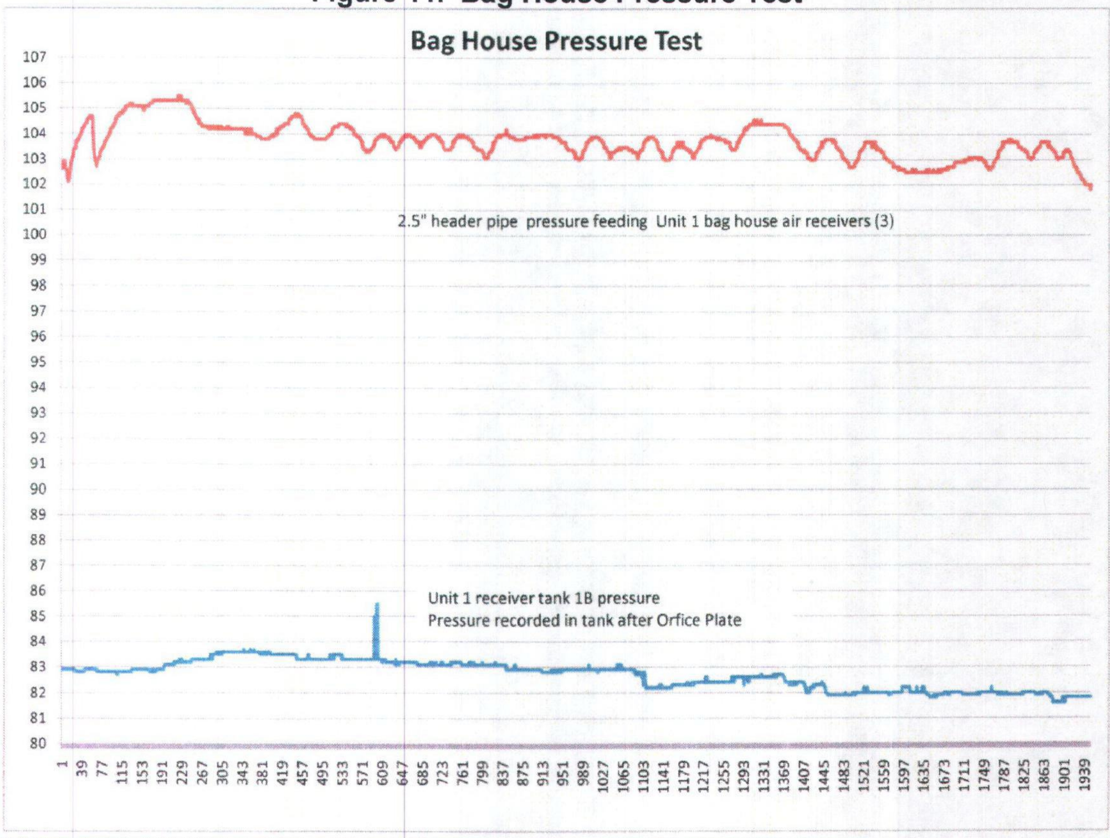


Figure 11. Bag House Pressure Test



IP12_001498

Figure 12. Unit #2 Bag House Cleaning Cycles

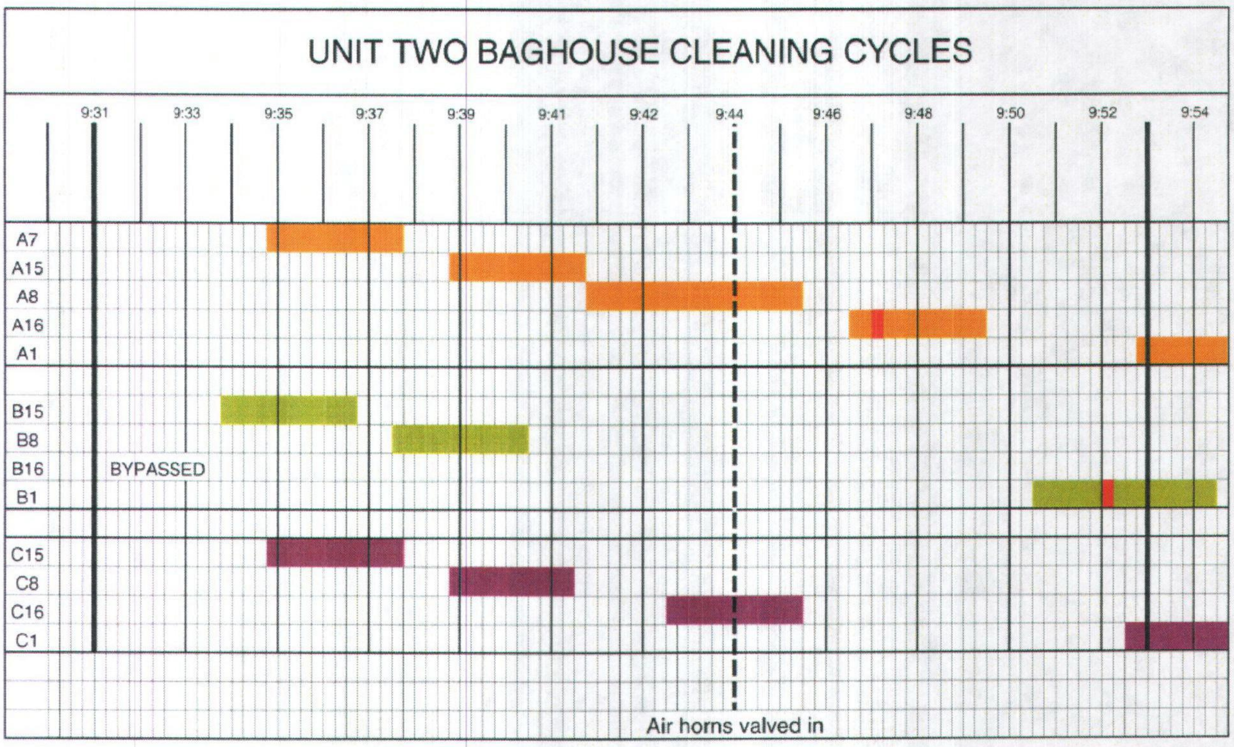
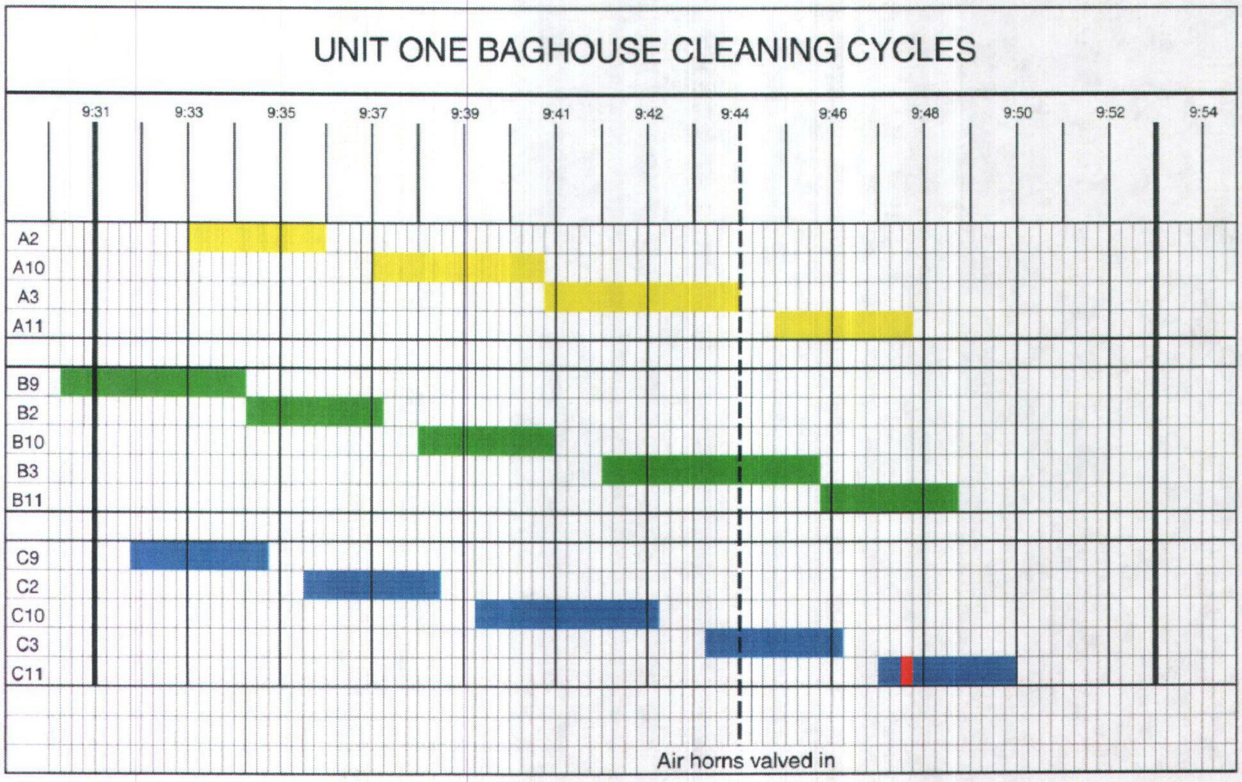


Figure 13. Unit #1 Bag House Cleaning Cycles



IP12_001499

Summary

Establishing the baseline for air demand flow and pressure for the Intermountain Power Service Corp. Electric Generating Station in Delta, Utah.

Measured demand as running today at 119 psig average pressure after the dryer discharge, reference the preceding line diagram and schematic:

Unit service flow average	1,365 scfm
Non-essential induce and outside air (average)	1,165 scfm
Units #1 and #2 Bag Houses average	2,864 scfm
Heatless dryer purge air average	1,335 scfm
Total (average)	6,729 scfm

Required system entry pressure is 100 psig.

Historical data shows at this similar operation. Last year, three compressors and four dryers ran 73% of the time and four compressors and four dryers ran the remainder of the time.

Controls – today's units operate with no turndown.

Operating Profile

The following chart titled "4 Compressors in Service" was supplied by plant personnel, which indicates that from March 9, 2007 to February 22, 2008, four units were on 27% of the time.

Four units operating hours (8,760 x .27)	2,365 hrs/yr
Three units operating hours (8,760 - 2,365)	6,395 hrs/yr

Current System Baseline

Tables 1 and 2 reflect the energy and economic performance of the current air system.

Table 1. Key Air System Characteristics – Current System*

Measure	Scenario A 3 Compressors and 4 Dryers on All Shifts	Scenario B 4 Compressors and 4 Dryers on All Shifts	Total
Average System Flow	6,729 cfm	7,259 cfm	NA
Avg Compressor Discharge Pressure	125 psig	125 psig	NA
Average System Pressure	105 psig	102 psig	NA
Input Electric Power	1,566.36 kW	2,088.48 kW	NA
Operating Hours of Air System	6,395 hrs	2,365 hrs	8,760 hrs
Specific Power	4.29 scfm/kW	3.4 7cfm/kW	3.96 scfm/kW
Electric Cost for Air /Unit of Flow	$74.43 /scfm year	$34.02 /scfm year	$108.45 /scfm year
Ann'l Elec Cost for Compressed Air (Main Year-round Production)	$500,844 /year	$246,963 /year	$747,807 /year
Summer Air Horn Use: (10 air horns x 124 scfm each x 1 kW per scfm x 2190 hours in summer season x 5¢ per kWh)	$27,156 /year		
TOTAL ANNUAL COST	$775,071 /year		

*Based on a blended electric rate of $0.05 per kWh, 8,760 hours/year.

Comments

The current compressors have local modulation control and are operating with no measurable turndown.

Table 2. Compressor Use Profile – Current System

Unit #	Compressor: Manufacturer/Model	Full Load		Actual Elec Demand		Actual Air Flow	
		Demand (kW)	Air Flow (scfm)	% of Full kW	Actual kW	% of Full Flow	Actual scfm
First Shift: Operating at 125 psig discharge pressure for 6395 hours							
1	3 Stage Elliott	522.12	2245	100	522.12	100	2245
2	3 Stage Elliott	522.12	2245	100	522.12	100	2245
3	3 Stage Elliott	522.12	2245	100	522.12	99	2239
4	3 Stage Elliott	522.12	2245	OFF			
			TOTAL (Actual):		**1566.36 kW**		**6729 scfm**
Second Shift: Operating at 125 psig discharge pressure and 2365 hours							
1	3 Stage Elliott	522.12	2245	100	522.12	100	2245
2	3 Stage Elliott	522.12	2245	100	522.12	100	2245
3	3 Stage Elliott	522.12	2245	100	522.12	99	2239
4	3 Stage Elliott	522.12	2245	100	522.12	23	524
			TOTAL (Actual):		**2088.48 kW**		**7259 scfm**

Current System Summary

Annual plant electric costs for air production, as running today, are $775,071 per year. These estimates are based on a blended electric rate of $0.05 /kWh.

The air system operates 8760 hours a year. The load profile or air demand of this system is relatively stable during all shifts

2.2 PROPOSED SYSTEM DESCRIPTION

The overall strategy for improving the air system centers on replacing the inefficient heatless dryers with blower purge combined with other air conservation programs to significantly reduce the amount of air demand. Reconfiguring piping where required to eliminate pressure losses.

Proposed System Changes

The specific projects to improve the air system are described in Chapters 3 and 4 of the report. Figure 2 provides a schematic of the proposed system. The recommended projects include:

Efficiency Projects

- Install central air management control system with appropriate inlet guide vanes and other auxiliary equipment, such as the CEC quotation we reviewed , overall unit turndown should go up to 80% average turndown and up to 80% power. — 3,118 scfm

Air Flow Reduction Projects (Total Reduction = 2794 scfm)

- Replace current heat of compression with blower purge units — 1348 scfm
- Replace current aftercooler gravity drains with appropriate level activated (8) — 110 scfm
- Reduce system pressure 19 psig to unit service air, non essential, inside, outside air — 481 scfm
- Repair tagged leaks, continue program — 855 scfm

Other Projects (Total Reduction = 208 kW)

- Run at least two units at/or near full turndown with the new air management control system. This is reflected in the proposed table by the effect of probable efficient turndown. — 208 kW, $91,104/ yr
- Replace air-operated air horns when used during summer season (2190 hrs) — 1240 scfm, $27,156

Proposed System Impacts

Tables 3 and 4 reflect the impact the proposed projects are expected to have on air system performance and operating costs of the current system reported in earlier Tables 1 and 2.

There are two categories of savings: savings reflected in comparing the compressor operating costs in the current system (Table 1) and the proposed system (Table 3), and additional savings not directly associated with operating the compressors, such as adding cycling refrigerated dryers.

Table 3. Key Air System Characteristics – Proposed System*

Measure	Scenario A 2 Compressors and 2 Dryers on All Shifts	Scenario B 2 Compressors and 2 Dryers on All Shifts	Total
Average System Flow	3935 scfm	4465 scfm	NA
Avg Compressor Discharge Pressure	110 psig	110 psig	NA
Average System Pressure	100 psig	100 psig	NA
Input Electric Power	856 kW	972 kW	NA
Operating Hours of Air System	6395 hrs	2365 hrs	8760 hrs
Specific Power	4.60 scfm/kW	4.59 scfm/kW	NA
Electric Cost for Air /Unit of Flow	$69.56 /scfm year	$25.74 /scfm year	$95.30 /cfm year
Ann'l Elec Cost for Compressed Air	$273,706 /year	$114,939 /year	$388,645 /year
Summer Air Horn Use: (no air horns x 124 scfm each x 1 kW per scfm x 2190 hours in summer season x 5¢ per kWh)	(Negligible additional electric use with electric-operated air horns – less than $1,000 per year)i		
TOTAL ANNUAL COST	$388,645 /year		

*Based on a blended electric rate of $0.05 per kWh, 8760 hours/year.

Table 4. Compressor Use Profile – Proposed System

Unit #	Compressor: Manufacturer/Model	Full Load		Actual Elec Demand		Actual Air Flow	
		Demand (kW)	Air Flow (scfm)	% of Full kW	Actual kW	% of Full Flow	Actual scfm
First Shift: Operating at 110 psig discharge pressure for 6395 hours							
1	3 Stage Elliott	522.12	2402	82%	428	82%	1968
2	3 Stage Elliott	522.12	2402	82%	428	82%	1967
3	3 Stage Elliott	522.12	2402	OFF			
4	3 Stage Elliott	522.12	2402	OFF			
		TOTAL (Actual):			**856 kW**		**3935 scfm**
Second Shift: Operating at 110 psig discharge pressure and 2565 hours							
1	3 Stage Elliott	522.12	2402	93%	486	93%	2233
2	3 Stage Elliott	522.12	2402	93%	486	93%	2232
3	3 Stage Elliott	522.12	2402	OFF			
4	3 Stage Elliott	522.12	2402	OFF			
		TOTAL (Actual):			**9724 kW**		**4465 scfm**

Proposed System Summary

The savings potential of the projects related to operating the compressors and a project of $1,212 per year to operate the new dryers total $385,214. Costs for implementing these projects still need to be quoted, but the total cost is expected to be less than $770K or a two-year payback.

Some of the key parameters characterizing the current and proposed systems and the associated savings projects are provided below.

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SYSTEM COMPARISON	CURRENT SYSTEM	PROPOSED SYSTEM
Average Flow (Scenario A / B)	6729 scfm 7259	3935 scfm 4465
Avg Compressor Discharge Pressure	125 psig 125	110 psig 110
Average System Pressure	105 psig 105	100 psig 100
Electric Cost per Cfm	$108.45 /scfm/yr	$95.30 /scfm/yr
Annual Electric Cost:		
Compressor Operation	$775,071	$388,645
Other Air Equipment	$*	$1,212 (new dryers)
Total Annual Electric Cost	$775,071	$389,857
OVERALL PROJECT EVALUATION:	**SAVINGS**	**COSTS**
Total	$385,714	< $770k (< 2-year payback)

*Note: The current dryers are heatless and have no direct energy cost but use 1348 scfm in purge air at $84.53 / scfm/yr which equals $113,946 per year. Replacing these is reflected in the overall compressor operating cost. The replacement heated blower purge dryers have a projected operating cost of $1,212 per year with two dryers running and the effective purge control.

IP12_001506

2.3 PROJECT EVALUATION METHODOLOGY

The specific supply-side and demand-side projects that form the basis of the new proposed compressed air system are described and evaluated in Chapters 3 and 4. In order to provide a reasonable value of the savings associated with each project, a methodology is used to allocate the total system savings among the individual projects. Such a methodology is motivated, in part, by seeking to avoid any potential double counting in savings estimates – a common mistake in many compressed air assessments.

The methodology is based on determining parameters for the "$ per psig saved" for pressure reduction projects, "$ per cfm saved" for flow reduction projects, and the "$ saved" for compressor efficiency and reconfiguration projects. Although any allocation approach can result in the savings parameter being set too high for one type of project (e.g., pressure reduction projects) and, correspondingly, too low for a second project type (e.g., flow reduction projects), summing the total savings for all the individual projects will match the total system cost improvement derived in Section 2.2.

In any case, it is always recommended that the entire set of recommended projects be implemented, because many of the projects are interactive in nature. Leaving out a single project could eliminate the effectiveness of the remaining projects that are implemented. Proposed air systems can also improve air quality, reduce maintenance costs, extend equipment life, reduce water use, improve environmental compliance, and reduce rental costs. For example, associated maintenance and other costs can often enhance project savings by 30% of the identified electric cost savings. If important to the assessment, these other savings can be tracked in addition to the electric cost reductions derived in Section 2.2.

Most of the overall program savings is simply the difference between the operating costs of the current compressors ($775,071 -- Table 1) and the proposed compressors ($388,645 -- Table 3) or $386,426. This figure needs to be decreased by $1,212 to $385,214 to reflect the direct electric use by the new drying system (Project #3).SED

Estimated savings from the supply-side projects (Projects #1 and #2) are derived in Section 3.1 and total $123,094 ($31,990 + $91,104).

This leaves $263,332 ($386,426 - $ $123,094) to be allocated among the air flow reduction projects (Projects #3 - #9). Of this amount, $27,156 is associated with the air horn project (Project #8) described in Section 4.5.1. The remaining of $236,176 ($263,332 - $27,156) is allocated among the air flow reduction projects, which are saving 2794 scfm at a calculated value of $84.53 per acfm.

IP12_001507

CHAPTER 3. SUPPLY-SIDE SYSTEM REVIEW

3.1 PRIMARY AIR COMPRESSOR SUPPLY

The primary air compressors are early 1980s technology, 3-stage, 100-psig class Elliott centrifugal compressors. There have been at least two generations of significant technological performance improvement since these units were produced. The currently available enhanced performance of this class of compressors has been a product of much more precise manufacturing capability and the ability of design engineers to improve upon previous design utilizing this more precise manufacturing.

New centrifugal compressors of this class will have a 10 to 20% better basic specific power. Intermountain Power has a proposal to add some specific product upgrades and accessories that will have a very significant positive impact on performance:

- Inlet guide vanes to allow full available turndown at a very favorable specific power.
- Target pressure controlled, full networking central air management system.

Installation and upgrading these units with proper auxiliary and accessory equipment will offset some of this inefficiency. Overall, the units appear to be well maintained and in generally good working order, except as noted.

Table 5. Comparison of Current and Proposed Compressor Ratings

Manufacturer	Elliott (#1)	Elliott (#2)	Elliott (#3)	Elliott (#4)
Model	310DA3	310DA3	D10DA3	310DA3
Unit Type	3-stage Cent	3-stage Cent	3-stage Cent	3-stage Cent
Type of Cooling	Water	Water	Water	Water
Full Load Nominal Published BHP	700	700	700	700
Full Load Horsepower (bhp) actual	660	660	660	660
Full Load Motor Efficiency (.me)/calc	.943/.90	.943/.90	.943/90	.943/.90
Full Load Pressure (psig)	108	108	108	108
Full Load Flow (icfm)	3,100	3,100	3,100	3,100
Full Load Flow (scfm)	2,245	2,245	2,245	2,245
Full Load Nominal Set Point (psig)	108	108	108	108
Type of Capacity Control	IBV/BOV	IBV/BOV	IBV/BOV	IBV/BOV
Pressure Control Band	125-135	125-135	125-135	125-135
Turn Down % (estimated avg)	11%	11%	11%	11%
Turn Down Air Flow (scfm)*	1,998	1,998	1,998	1,998
Full Load (input) kW @ 108 psig: Calculated	522.12	522.12	522.12	522.12
Turn Down kW: 11% (5% power)	496.02	496.02	496.02	496.02
Idle kW (estimated)	183	183	183	183
Full Load Specific Power (scfm/kW)	4.834	4.834	4.834	4,834
Annual Electric Cost ($/scfm)*	$121.05	$121.05	$121.05	$121.05

* Based on blended electric rates of $0.05 per kWh and operation of 8,760 hours per year.

**Plant uses 2,245 scfm each for capacity.

acfm to scfm multiplier = x **.7315**

$$100 \times \frac{(12.2\text{ psia} - .9492\text{ psia})}{14.5\text{ psia}} \times \frac{528°F}{460 + 100°F} = \frac{528}{560}$$

$$100 \times \frac{(11.25)}{14.5} \times \frac{528}{560} = .7315$$

Scfm calculated:
68°F / 14.5 psia / 0% RH

Operating Condition:
Ambient Temp100°F used/20% RH

$$kW = \frac{amp \times volts \times 1.732 \times PF}{1{,}000}$$

$$522.12 = \frac{(amps)\ 6600 \times 1.732 \times .90}{1{,}000}$$

Full Load: Estimated amps at 6600 volts = 50.75 kW
Estimated full turndown amps at 11% TD = 48.21 amps

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☑ **RECOMMENDED PROJECT (#1)** – Reduce compressor discharge pressure from 125 psig to 110 psig after the piping has been reconfigured.

This will increase the design air flow 7%, but operating at about the same power, and extend the turndown range. Standard 125 psig flow – 2,245 scfm plus 7% extra air from each operating unit about the same energy use.

Each compressor can now deliver 2,402 scfm each (2,245 x 1.07) for a gain of 157 scfm each or 628 scfm for four operating units. Because the proposed system is expected to run with just two units turned on, the estimated actual air flow increase with two units on is 314 scfm.

Net additional air (2 units @ 157 scfm increase each)	314 scfm
Value of additional air -- average cost of air between current system ($ 108.45 /scfm per year in Table 1) and the proposed system ($95.30 /scfm per year in Table 3)	$101.88 /scfm yr
Total value of additional air	$31,874

Project costs – included in Projects #2 and #6

IP12_001510

3.2 COMPRESSOR CAPACITY CONTROL

The two most effective ways to run air compressors are at "Full Load" and "Off."

Capacity controls are methods of restricting the output air flow delivered to the system while the unit is running. This is always a compromise and is never as efficient as full load on a specific power (cfm/hp) basis. For details on unloading, see the MISCELLANEOUS SECTION in the back of this report.

Centrifugal Controls

The two most common controls used for centrifugal compressors are **modulation** and **blow off**. Modulation is relatively efficient at very high loads, but will not work much below 70-75% load. The four units at Intermountain Power Delta have 11% design turndown on three units and 15% on the other unit. After "modulation" or "turn-down", the compressor will then just "blow off" excess air. The basic power draw at the blow off point will stay the same regardless of the load. The actual operating turndown of the units as installed is really very low – apparently less than 5%. With the current type of inlet butterfly valve operators find it difficult, if not impossible, to securely avoid "surge" due to the active turbulence as the valve closes. The net result is little, if any, actual turndown. More importantly, IGVs make it routine to be able to take advantage of the full turndown. There are many times because centrifugals are a mass flow-type compressor when atmospheric conditions will allow greater than designed turndown.

A modern, well-applied electronic air management system, combined with effective inlet guide vanes will greatly enhance the operating efficiencies of these particular compressors.

For more information on inlet guide vanes, see the MISCELLANEOUS SECTION – article reprint form Plant Services entitled "Control the Air."

Installing a management control system will allow the average of up to 20% turndown at 80% inlet power versus no turndown and 100% of power currently. This is a minimum savings of 104 kW per unit at part load or an average of two units at most times or 208 kW.

There are many other reasons to implement a professional compressed air management system including:

- Efficient, effective, and timely response.
- Improved and lower cost predictive and preventative maintenance.
- Allows proper performance checks
- Enhanced reliability and documentation of unscheduled downtime.

However, in all likelihood, this system will also probably reduce the overall energy operating electric energy cost at the compressor motor input.

☑ **RECOMMENDED PROJECT (#2)** – Install a new, modern electronic compressed air management control system combined with individual unit inlet guide vanes, replacing the current inlet butterfly valve system.

Net projected average kW reduction (running projected two units)	208 kW
Annual electrical energy cost ($0.05 kW @ 8,760 hrs/year)	$91,104
Estimated cost of project	TBD

3.3 AIR TREATMENT AND AIR QUALITY

3.3.1 Dryers

Desiccant dryer equipment removes moisture vapor by "adsorbing" it to desiccant beads (see MISCELLANEOUS SECTION). These dryers can consistently deliver a pressure dew point to -40°F or lower, which means they will remove more water vapor than refrigeration units. They regenerate the wet tower, while the other tower is drying. This requires the use of some type of heat and dry air to "sweep" or "purge" the exchanged moisture out.

The most common type of desiccant dryer is a twin tower, regenerative, desiccant dryer. These are most capable of delivering a consistent nominal –40° pressure dew point at rated scfm flow and purge when:

- Air is delivered to the dryer at less than 100°F
- Air is delivered to the dryer at no less than 100 psig
- Ambient air temperature is no more than 100°F
- The condensate is driven out of the aftercooler, pre-filter, and dryer is immediately removed from the system and is not allowed to re-entrain or build up
- No liquid water enters the dryer
- The dryer is not overloaded in volume (scfm)
- Air is not re-contaminated by moisture "wicking" past air leaks in the system
- Inlet air at 130°F or more will not be dried at all.

Water or Oil Carryover in System

Water (condensate) and oil carryover problems in the current air system are not significant when the dryers are working. The current dryers are almost 25 years old and utilize some older-style valves and controls. Currently, they have not shown a high degree of reliability according to plant personnel. Any problems can usually be expected to increase in magnitude during more humid months. The correct way to eliminate water and oil in the air system is to clean and dry the air immediately after it is produced in the compressor room. Then clean dry air can be stored in a separate air receiver and can flow to the system, as required. Some guidelines include:

1. Generally, it is best to eliminate water/oil at the air source before they enter air system.
2. Water vapor, when condensed to liquid in the drying process, must be removed immediately or it can recontaminate compressed air by evaporation and overflow.
3. Every 20°F increase in temperature will almost double the "moisture load" that air will hold. Compressed air dryers are usually capacity rated at 100°F and 100 psig inlet air conditions. At 120°F and 100 psig, the dryer's capacity rating is reduced by 50%.
4. Putting dry/oil-free air into the system 90% of the time and then allowing wet/oily air to enter sporadically 10% of the time will, in reality, make the system wet all the time.

The water and/or oil will fall out in the piping system and continue to re-entrain and contaminate and/or collect in the "low spots" of the system. This will cause recontamination as liquid is pulled into the flowing compressed air system. Bypassing the dryer with "part of the air" (controlled by the bypass valve) will almost always end up with "wet air"! A wet system could take many months of continued flow of clean dry air in order to "clean up."

5. It is best to identify required pressure dew point and meet it. Performance should be monitored closely, if critical. Intermountain Power Delta basically requires instrument-quality compressed air able to handle control valves and controls under winter conditions. It should be pointed out that extremely cold air holds insignificant volumes of water vapor. Many winter problems with "freeze up" come from condensate build up in low spots during more humid times, which either re-entrains into the system or actually freezes in a critical spot.

Current Drying System

Key features of the plant's current dryers are displayed in Table 6, along with the key features of proposed dryers recommended in this report.

Table 6. Comparison of Current and Proposed Dryers

Manufacturer	Current		Proposed	
	Pall Trinity	**Total of 4 Units**	**Desiccant**	**Total 4 Units**
Model	Desiccant	4 ??	Desiccant	--
Unit Type	Heatless	Heatless	Heated blower purge	Heated blower purge
Rated Flow @ 100°F/100 psig	2,242		2,242	--
Purge: scfm (CompAir)	337	1,348	NA	NA
Full Load Heater kW	NA	NA	36x.75=27 avg kW	108 avg kW
Full Load Blower hp/kW	NA	NA	15/12.5	50 kW
Total kW	NA	NA	39.5	158
% Load w/ Dew Point Demand Control	100%	100%	35%*	35%
Net Purge	337	1,348	13.83	55.32
Total Annual Cost ($)	**$28,487 /year**	**$113,946 /year**	**$605.75**	**$2,423 /year**

Based on blended electric rates of $0.05 per kWh and operation of 8,760 hours per year.

*Proposed dryers equipped with dewpoint demand controller.

☑ **RECOMMENDED PROJECT (#3)** – Replace four current heatless dryers with blower purge-type of similar rating with automatic dewpoint demand controllers. Eliminate 1,348 current purge blow off.

Total annual operating cost of current dryer (1,348 scfm) [Table 6]	$113,946 /yr
Total annual operating cost of proposed dryer (for two compressors) [Table 6]	$1,212 /yr
Total annual electrical energy cost savings	$112,734 /yr

Regeneration is accomplished by external heater and blower purge flows and the proposed dryer will be equipped with appropriate purge controls.

Air Suitable for Breathing

If the application calls for purified air for facemasks, hoods, helmets, and other supplied-air breathing apparatus, you may need a breathing air system. There are complete utilized, purification systems designed to remove excessive moisture, solid particulates (dust and dirt), oil and oil vapor, carbon monoxide, and other hydrocarbon vapors commonly found in ordinary compressed air. Air flows through a breathing air system, including a number of filter-purifying stages, and a catalyst to covert carbon monoxide to carbon dioxide. Various contaminants are removed at each stage until final "Grade D" level air is produced, which is air suitable for breathing under OSHA standards.

The following table outlines the basic OSHA and Canadian limitation on breathing air purity. Note the following regarding pressure dew point.

- Moisture dew point temperature – 10°F below ambient temperature (@ 1 atmospheric pressure)
- Dryness <u>not to exceed</u> (-)50°F at 1 atmospheric pressure.

Many operators believe that too dry air [below (+)10°F PDP] will make breathing uncomfortable due to excessive dryness. In any event, to meet the maximum limitation 10°F below ambient temperature at 1 atmospheric pressure means that a (+)40°F pressure dew point at 100 psig would be a (-)10°F dew point at 1 atmospheric pressure and would be acceptable to ambient or as low as 0°F.

Grade D – Breathing Air*

Contaminant	OSHA (Resp. Prot 1910.134)	CSA	Outlet Concentration at Rated Conditions
Oxygen (%)	19.5 to 23.5	20 to 22	
Carbon Monoxide	10 ppm	5 mL/m^3	10 with a max inlet concentration of 135:5 with max inlet condition of 100
Carbon Dioxide	1000 ppm	500 mL/m^3	CO is converted to CO_2; although some CO_2 is adsorbed in the desiccant beds, high concentration of CO_2 at the compressor intake, in addition to the CO_2 produced by the purifier
Oil and Condensate Hydrocarbons	6.65 ppm 5 (mg/m^3)	1 (mgL/m^3)	0
Odor	Lack of noticeable odor	Free of any detectable odor	None: purifier will remove gases contaminants normally removed by carbon
Moisture Content Dew Point Temperature	10°F (5.6°C) below ambient temperature (at 1 atm pressure)	9°F (5°C) below the min temperature breathing air is exposed (at line pressure)	Does not exceed –50°F (-45.6°C) when purified @ 100 psig and reduced to 1 atm pressure

*Contaminant and maximum allowable limit required by OSHA in the U.S. and Canada (OSHA 1910.134(i)(1) (ii) (Table 1).

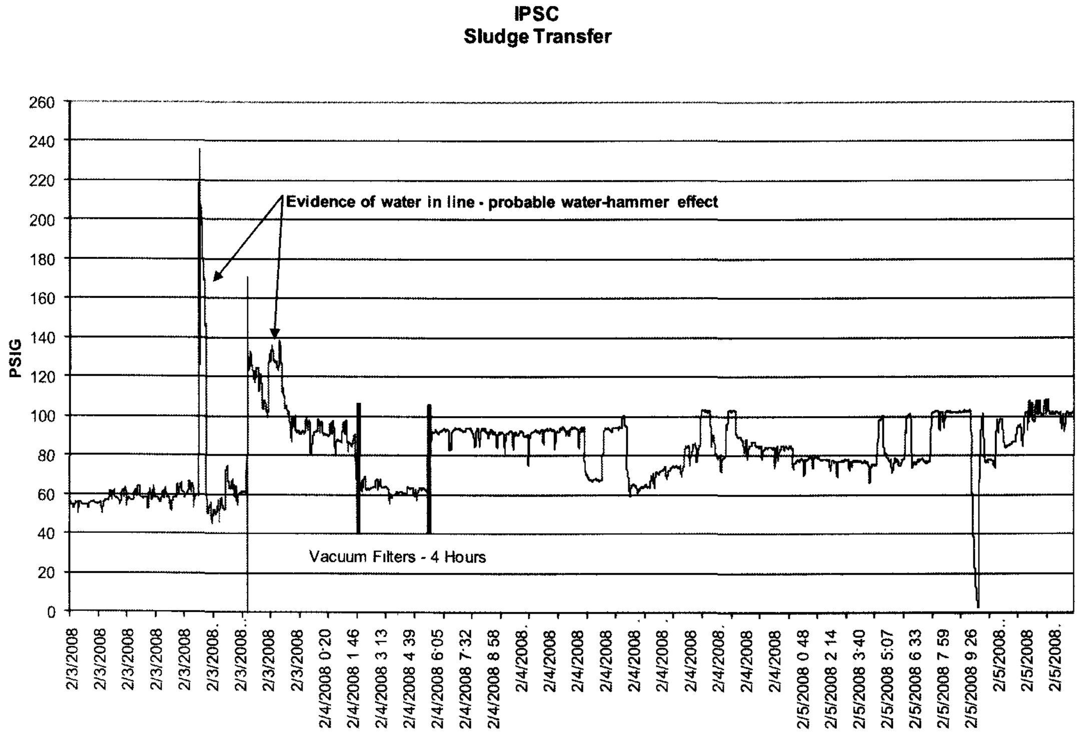
Aftercoolers

Aftercoolers are water cooled and currently appear capable of delivering 100°F or lower temperature compressed air to the dryer. If the aftercooler is not performing correctly, then the dryers may be undersized and operating ineffectively, creating poor air quality.

Some key data on this:

- 20°F rise in inlet temperature above the rated temperature (normally 100°F) will double the moisture load on the dryer or reduce the dryer's capability by 50%
- Above 130°F, desiccant dryers will not dry at all. All water vapor goes downstream
- Liquid water entering a desiccant dryer will not be removed by adsorption to the beads – it will either go on through or react with the desiccant dust to plug the dryer and foul the element
- Liquid water going into refrigerated dryer water will suck off all the refrigeration due to the high latent heat. The water vapor may not be condensed and pass into the system as vapor to condense later on downstream
- Condensate/water will always run to gravity – for example, on the inside of the pipe
- Water vapor will always flow from a higher relative humidity to a lower, regardless of the air flow.

Figure 14. Sludge Transfer



Summary

The complete air system is wet throughout all the headers and this water is giving the plant problems. There is rust and scale in the pipes, valves, etc. Water slugs or "water hammers" are spiking the pressure. Frozen water is blocking the lines and valves.

There are many open drains left cracked open to bleed water because of the problems.

The probable cause for this is the opening of the bypass valves around the dryers for whatever reasons. When wet air goes into the system, the system quickly becomes wet. We believe there are currently significant volumes of water stored in low spots, tanks, etc. This must be drained and/or evaporated out before the total system can become dry.

Once the new dryers are installed or the current dryers are rebuilt to a higher degree of reliability (which may not be possible due to age and obsolescence), due diligence to operation and maintenance combined with the success of Projects #6 and #7 should create an atmosphere and program to preclude this happening again.

Projects #6 and #7, if implemented successfully, will eliminate the excessive piping pressure loss and eliminate any reason to bypass the dryers.

Compressed air reduction projects should leave you with one swing/back-up compressor and one swing/back-up dryer, which should allow proper shutoffs and maintenance.

During the "drying up" process, the system may well create some possible significant accumulations of rust and scale for about 4 to 6 months, which will have to be addressed.

3.3.2 *Condensate Drains and Handling*

Background

Automatic drain traps come in three categories: Level-operated mechanically activated, dual-timer electronic, and level-operated electronic drains.

Level-Operated Mechanically Activated Drains. These drains do not waste air, but are prone to clogging and require continuing maintenance to assure operation. These drains work best in a "Power House Situation" where regular attention on an ongoing basis is part of the operation. Drain prices range from $65.00 each to $250.00 each.

Dual-Timer Electronic Drains. These drains use an electronic timer to control the number of times per hour it opens and the duration of the opening. The theory is that the times should be adjusted to be sure that the condensate drains fully and the open time without water is minimized, because it wastes compressed air. The reality is that the cycles often don't get reset from the original factory settings. This results in condensate build-up during the summer and in getting set wide open and not closed down later during in cooler weather. When they fail "stuck open", they blow at a full flow rate of about 100 cfm.

Consider, for example, that the usual "factory setting" is 10 minutes with a 20-second duration. Some 1500 scfm of compressed air will generate about 63 gallons of condensate a day in average weather or 2.63 gallons per hour. Each 10-minute cycle will have 0.44 gallons to discharge. This will blow through a ¼-inch valve at 100 psig in approximately 1.37 seconds. Compressed air will then blow for 18.63 seconds each cycle, 6 cycles a minute, which will total 111.78 seconds per hour of flow or 1.86 minutes per hour of flow. A 1/8-inch valve will pass about 100 cfm. The total flow will be $100 \times 1.86 = 186$ cubic feet per hour, or $186 \div 60$ minutes = 3.1 cu ft/min on average. This 3.5 cfm would translate into an energy cost of $300 per year based on a typical air flow cost of $100 per cfm year.

Depending on the type of discharge valve (whether it is solenoid-operated or motorized ball valve-operated and whether the timer is dual-type with test button or remote alarm), the valve prices range from $89 to $600 each.

Level-Operated Electronic and Pneumatic Drains. These drains come in a number of varieties, including ones that receive the signal to open from a condensate high level and the signal to close from a condensate low level. These drains waste no air and are the best selection from a power cost standpoint. Their reliability is usually many times greater than the level operated mechanical drains. Prices range from $250 to $850 for standard products (more for specials).

Be sure auto drains are set up to work effectively. Some guidelines include making sure all drains:

- Are not tied together to a common header
- Can be checked easily for operation
- Are properly "vented" to atmosphere, if necessary
- Are sized, piped "to" and "from" with the full capability to handle anticipated highest humidity weather loads

- Have a bypass bleed on the feed pipe
- Can be easily checked, if they are passing condensate.

Connect each drain's point (after-cooler, pre-filter, dryer, after-filter, receivers, and all risers) separately to individual level-activated electric or pneumatic drains to collect and direct the condensate to a proper handling point carry it in a large plastic vented line (4" or 6"). Be sure maintenance personnel can effectively and visually monitor the drain's action.

Current Application

The configuration and performance of the condensate drains in the plant's compressor area do need to be modified.

☑ **RECOMMENDED PROJECT (#4)** – The condensate drains from the aftercooler separators on the four primary compressors are valved to drain continuously. During our site visit, we noticed the aftercoolers from Unit 1D and 1C were both blowing significant amounts of compressed air continuously. The total lost air for all four compressors as observed is 110 cfm. We recommend the plant install two level-operated pneumatic-actuated drains rated at maximum handling capacity of 10-12 gph at each aftercooler dryer. Tie these together with a "Y" connection, 1" pipe to and from each drain to your current collector pipe. Also install one drain in the riser to the dryers.

Total of number of drains	8 pts/16 drains + 1
Total compressed air saved	110 cfm
Recoverable energy savings from air flow reduction [Section 2.3]	$84.53 /cfm yr
Total annual energy savings	$9,298 /yr
Cost per drain (materials and installation) – 17 drains for aftercooler/ 1 for riser)	$550 each
Cost of project (17 drains)	$9,350

Figure 15. Level-Operated Drain on All Wet Side Risers and Drain Points

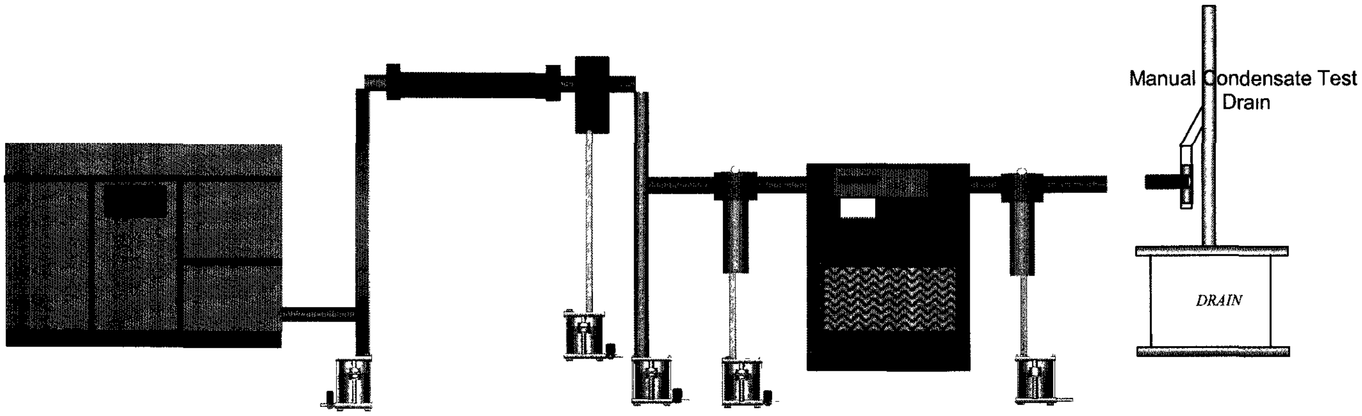
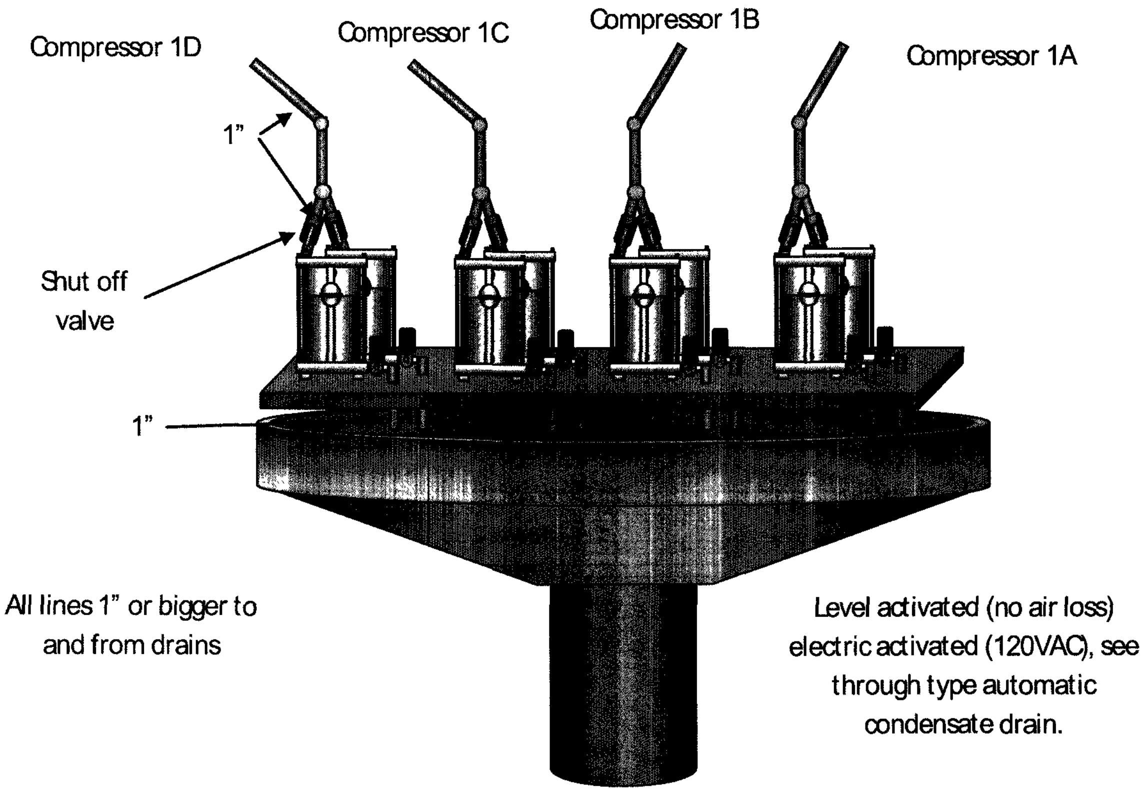


Figure 16. Compressor Aftercooler Drain Suggested Changes



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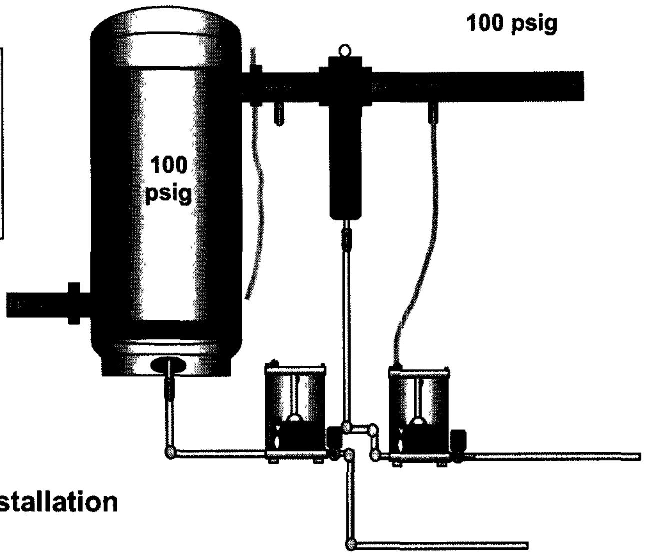
Don't "Air Lock"

Drain "Air Locked"

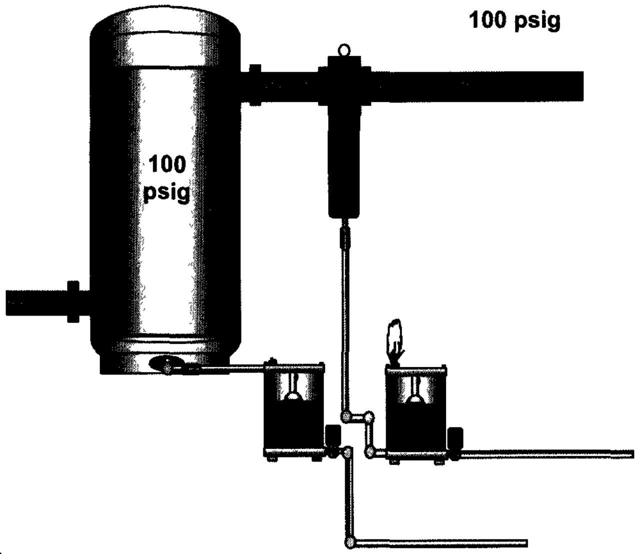
- Pressure equal – drains air lock – no additional condensate can come in
- Drains open – condensate stays in filter and/or air receiver



Recommended Installation

Recommendations to Avoid "Air Lock"

- Top feed into drain – no vent needed
- Vent line open to atmosphere with bottom feed



Compressed Air Condensate Handling

Plant personnel state that the condensate goes to water treatment. If this is true, and if discharge condensate meets the requirements of the local water treatment facility, there is no problem. (Refer to the Article Reprint – "Do You Know Where Your Condensate Is?" in the MISCELLANEOUS SECTION.)

However, if the plant is discharging the condensate to a storm sewer or to some other ground water, the plant may be required to separate it by the local water treatment facility—Federal EPA minimum is 10 ppm. While there is no significant energy impact associated with most condensate handling projects, such projects can be critical to achieve environmental compliance and avoid environmental penalties. Cost for condensate-handling systems often fall in the $3-10K range.

The compressed air condensate handling process at all plants should be reviewed to ensure environmental compliance.

IP12_001521

CHAPTER 4. DEMAND-SIDE SYSTEM REVIEW

4.1 BASIC SYSTEM HEADER AND PIPING

Background

It is the job of the main header system to deliver compressed air from the compressor area to all sectors of the plant, with little or no pressure loss. The header should be checked at appropriate points with a single test gauge. If there is a significant pressure drop anywhere, then corrective actions are likely needed. A pressure loss of no more than 1 to 3 psid is a reasonable target.

It is also desirable that the compressed air velocity in the main headers be kept below 20 fps to allow effective drop out of contaminants and to minimize pressure losses caused by excessive turbulence. The magnitude of the turbulence effect depends on the piping layout and pipe size.

Typical header projects include adding pipe, replacing pipe with larger diameters by adding angled connectors, and re-orienting or re-directing air flows.

Investigation of the Bag House Compressed Air Usage – Units #1 and #2 and the Overall Air Distribution Supply

Units #1 and #2 Bag House utilize reverse flow. Following specifications:

- Each unit has 3 casings
- Each casing has 16 compartments
- Each compartment has
- 2 inlets
- 2 outlets
- 2 reverse air
- purge air (fan air)
- air horns
- 2 vibrators.

Each casing has:

- 4 pre-casing cylinders (14"x66")
- 1 relief cylinder (5"x22")

Trended plant data has each compartment running 1 cycle every 24 hours and using about 2,646 cfm or 482 cfm per casing per Bag House.

- air horns (every third cycle) used to flutter the bags
- 2 air vibrators per component

The flow to these two Bag Houses is controlled by orifice plates. Originally, there were a total of six orifices installed on each of the sets of three air receivers feeding the compressed air to each Bag House. These are .421" ID rated to flow 298 scfm each at 100 psig, 70°F inlet condition. Estimated total anticipated flow is 1,800 scfm.

Three years ago, larger orifice plates were installed on the 2" entry lines to the receivers for Unit #2. These new orifice plates have an ID of .621" and are still in use. These are rated to flow 630 scfm each at the same inlet conditions.

New total anticipated flow:	3 @ 298	894
	3 @ 630	1,890
		2,784 scfm

Other Changes

These bag houses have had to handle more product with new coal supplies and the cycle time was increased, which would increase some of the air cylinder use.

Air horns (venturi air flow) were added to the process in order to increase the material handling capability with optimum bag life. Each air horn is rated to use 75 scfm at 75 psig (estimated 94 cfm @ 100 psig) entry pressure and moving 2,400 scfm with air horns to improve the performance of the reserve flow.

Three air horns operate every third cycle of five minutes and for 80-90 minutes per compartment. As the air horns go through the operating cycle, they are computer-controlled and blow for ten seconds (average flow of 12.5 scfm/rate of flow – 75 scfm @ 75 psig inlet pressure) every 4.5 minutes. They only operate in each compartment every third cycle. Their effect on the dynamics of the air system at this time appears to be insignificant.

Current Application

The current air distribution system was monitored throughout the audit process a 19 different points where calibrated pressure transducers with data loggers were mounted. The system sketch (pp.38-40) reflects the locations and average pressure readings during production as observed (for a complete set of trended downstream pressure profiles, see the PLANT SURVEY Section of this report).

Overall, the distribution system seems to be very adequate with a few notable exceptions as listed. Following are some of the significant areas where pressure loss is probably having a very negative impact on energy cost, reliability, and productivity:

Feeds from the main header to Units #1 and #2 Bag Houses. As described earlier in the baseline section, the air flow to the three receivers before the casings at each unit flow through restrictor orifice plates installed on the inlet to each receiver.

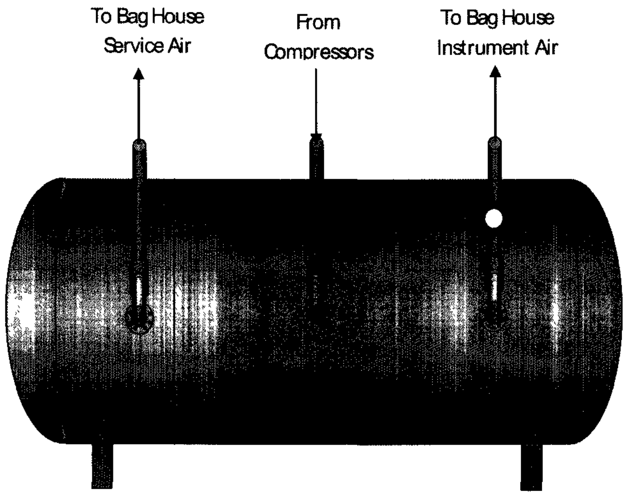
These are resulting in 20 psig loss to the Unit #1 Bag House (smallest orifice plate) and 10 psig to the Unit #2 Bag House (larger orifice plate).

Plant personnel state that the optimum operating pressure for the Bag Houses is 80 psig, which they cannot currently hold now in a continuing manner.

When all four dryers are on, the total pressure loss will probably make the sustainable entry pressure to the Bag Houses between 60 and 70 psig. The orifice plates were probably installed to sop an uncontrolled Bag House from pulling down other critical pressure areas.

☑ **RECOMMENDED PROJECT (#5)** – Remove orifice plates on the entry to each of the six air receivers. Install appropriately sized regulator recommend sized to handle 1,200 to 1,500 scfm with an 85 psig steady entry pressure and 80 psig delivered.

Figure 17. Tank Recommended Changes



Install these on the discharge line from each receiver to each casing. This will control the Bag House and protect other areas and allow better use of the stored air in the receivers.

There is a 16 psig loss in pressure from the dryer discharge to the main distribution header. This, of course, has a negative impact on all the following systems.

This is not an energy issue, but will allow better and more reliable operation.

☑ **RECOMMENDED PROJECT (#6)** – Reduce and control the main header entry pressure to 100 psig from 119 psig after re-piping and elimination of 16 psig loss from the dryer to the header.

Current flow of unit service air and non-essential indoor/outdoor air at 119 psig entry pressure at header (measured) = 2,530 scfm.

Projected air flow at 100 psig compressed air savings (2,530 x .81)	2,049 scfm
Compressed air savings (part of re-piping project)	481 scfm

For details, see the following schematics.

Annual electrical energy cost per scfm per year	$84.53/scfm/yr

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Total scfm year saved (running two proposed units)	481 scfm
Total recoverable annual electric energy savings	$40,659 /year

Figure 18. Typical Compressor Arrangement

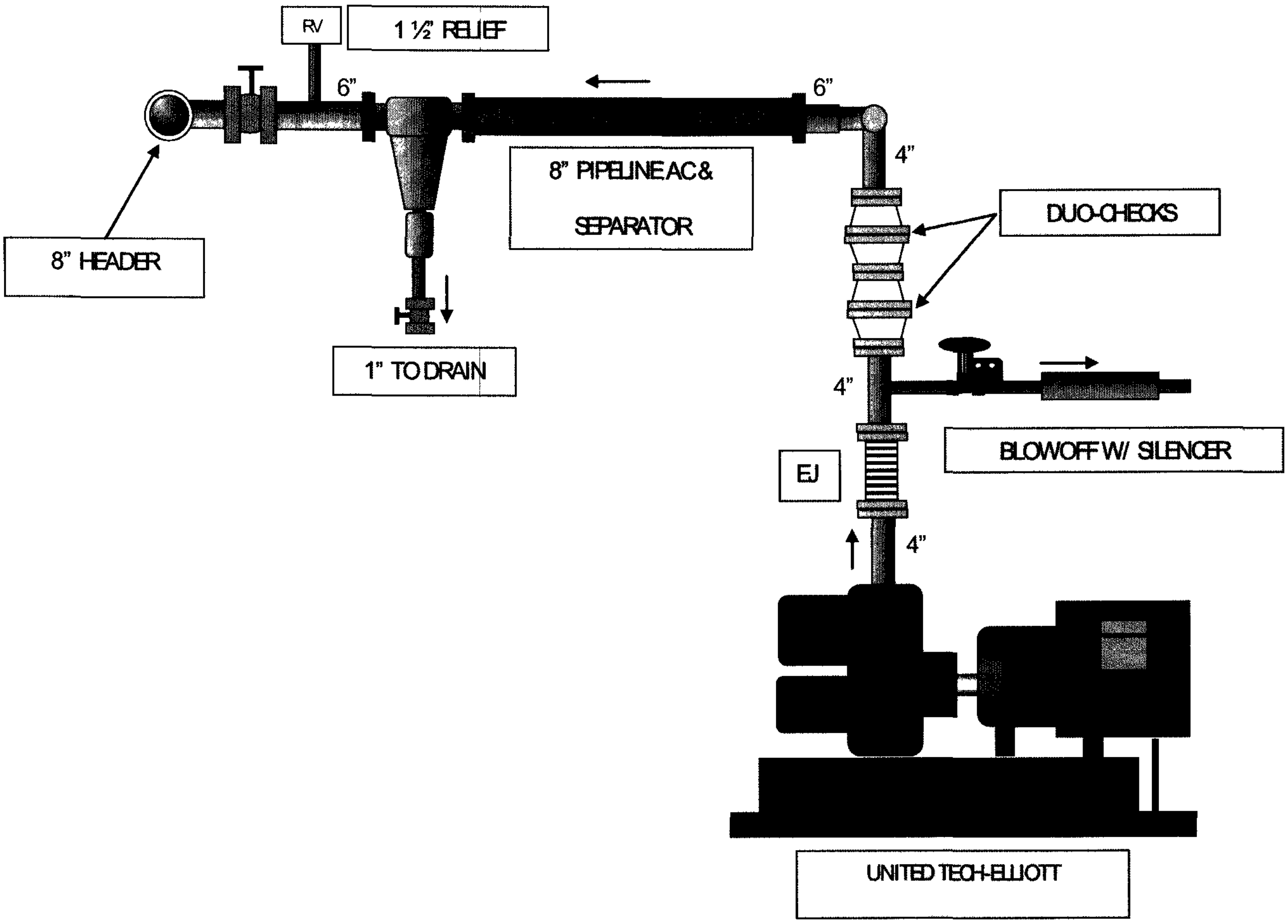

Figure 19. Compressor Floor Piping

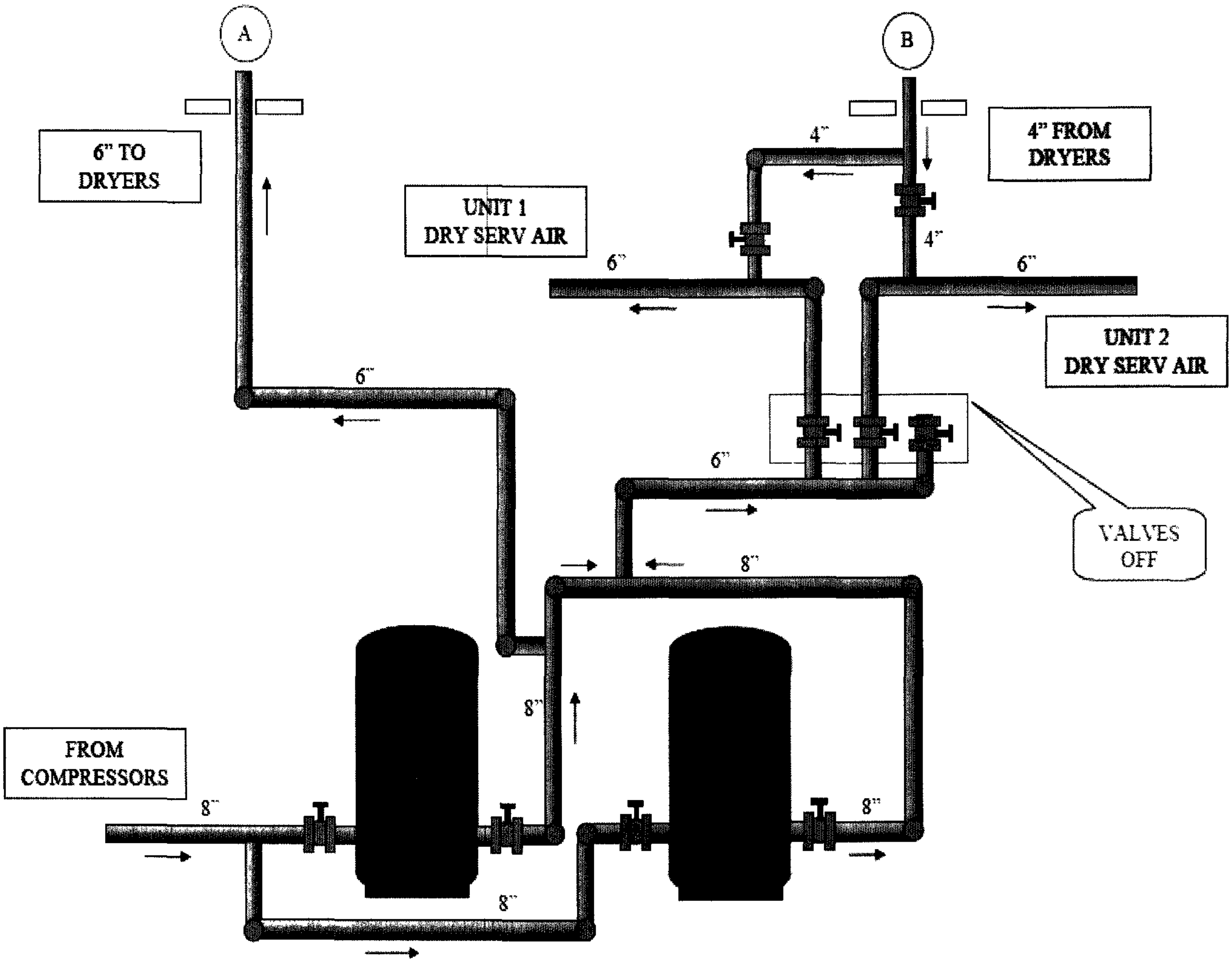

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Figure 20. Dryer Floor

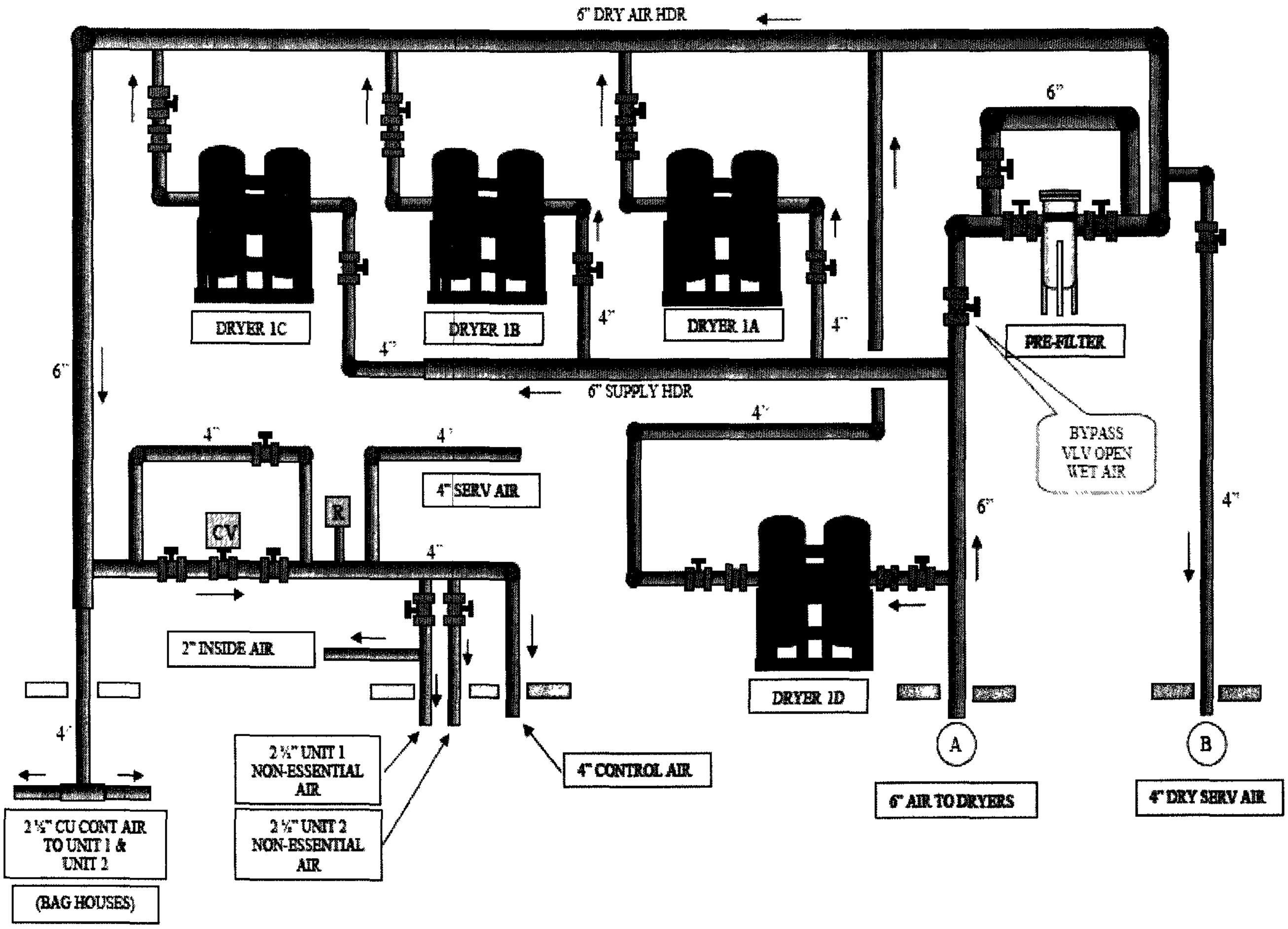

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Figure 21. Pressure Readings for Control Air Main Header

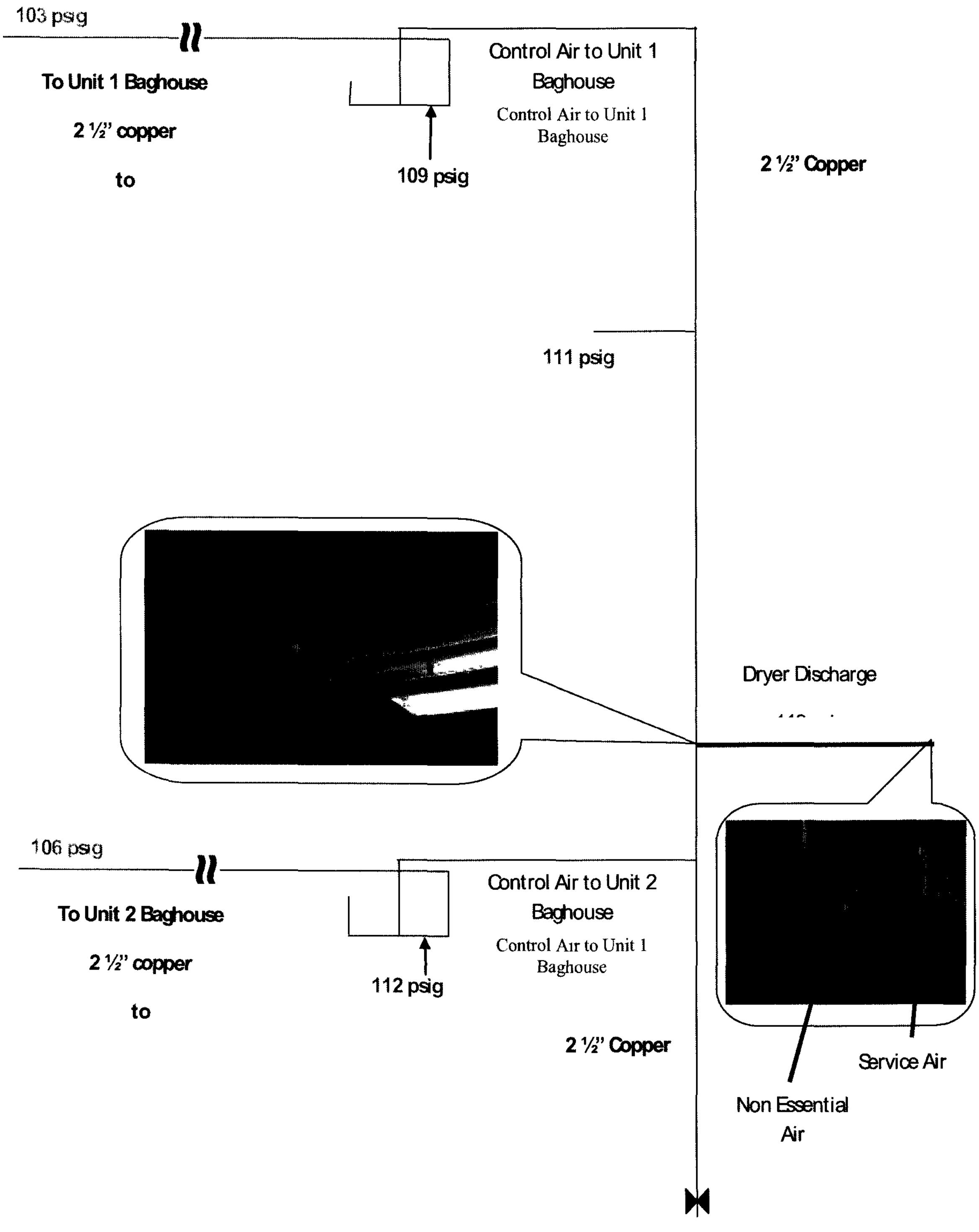

Figure 22. Compressor Receiver

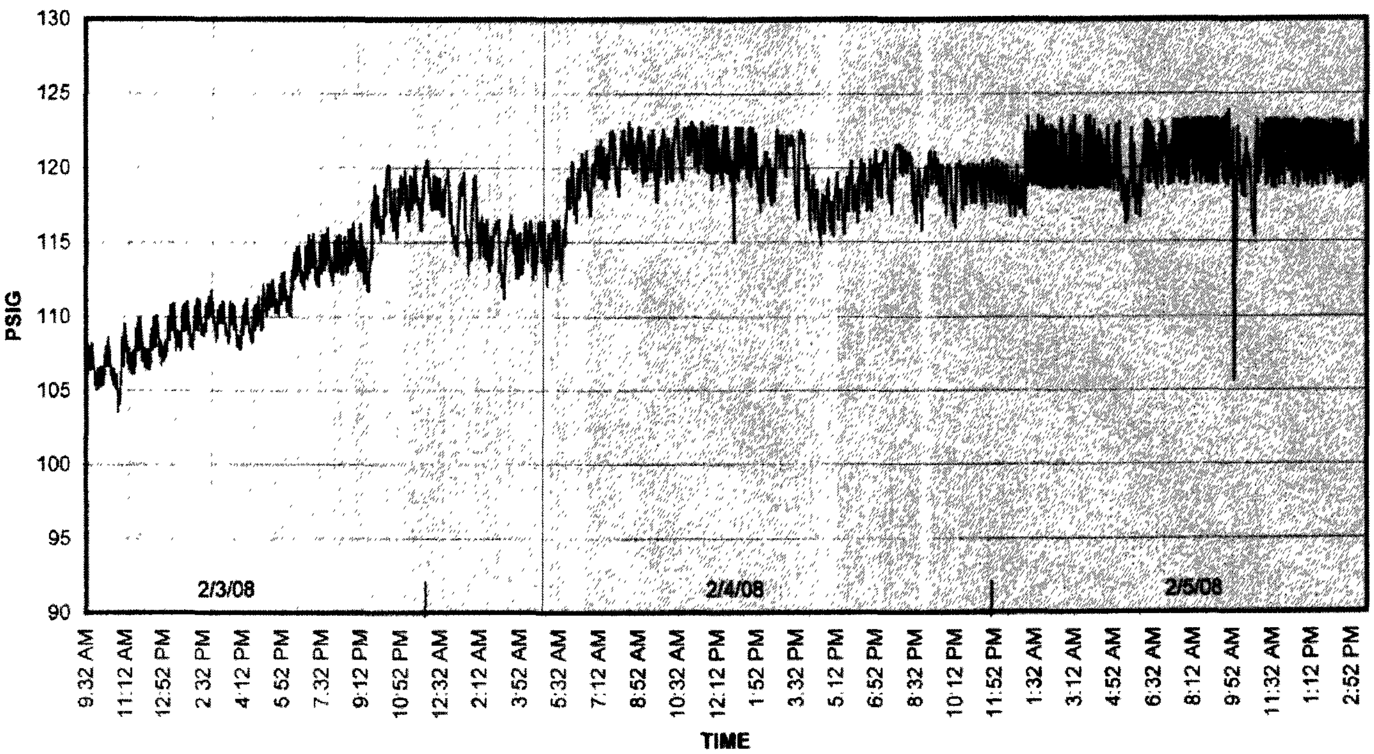

Figure 23. Unit #1 Scrubber Receiver

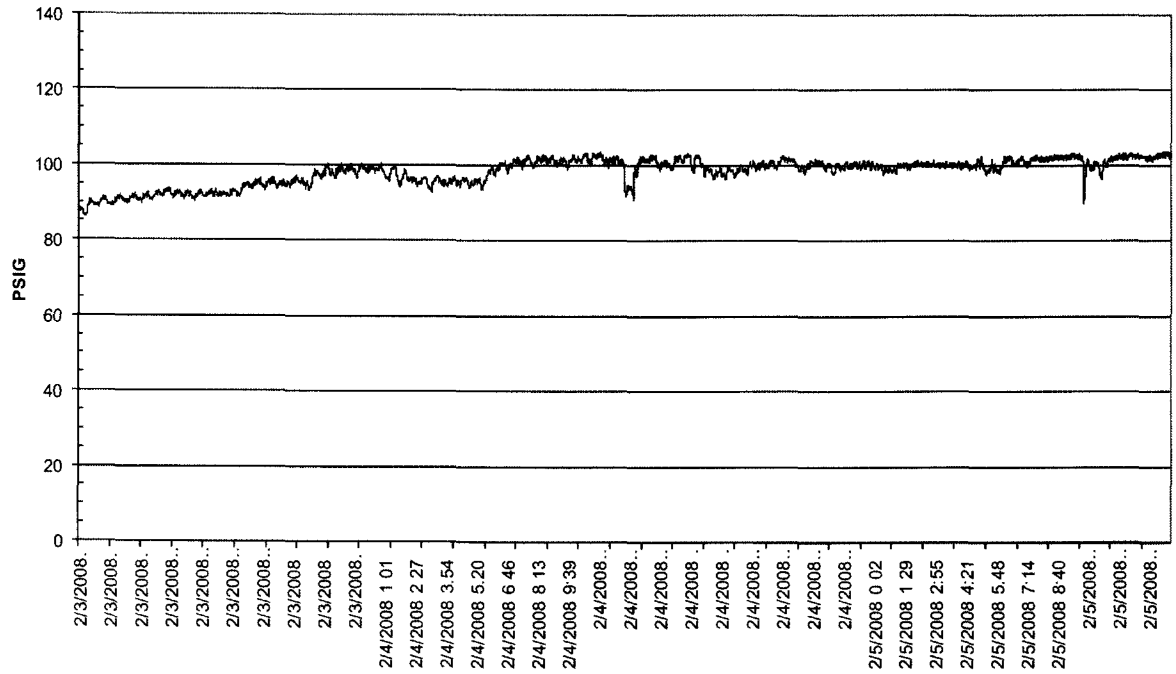

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Figure 24. Unit #2 Scrubber Receiver

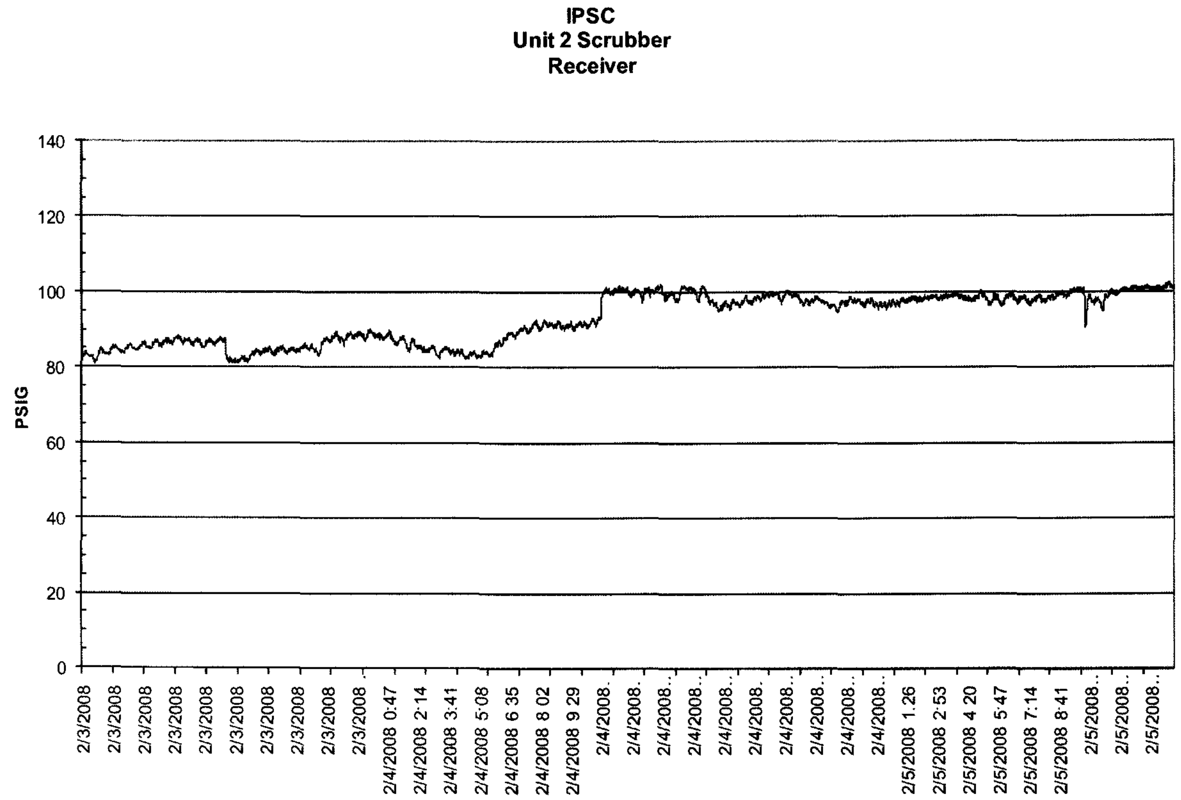

Figure 25. Transfer Building 4

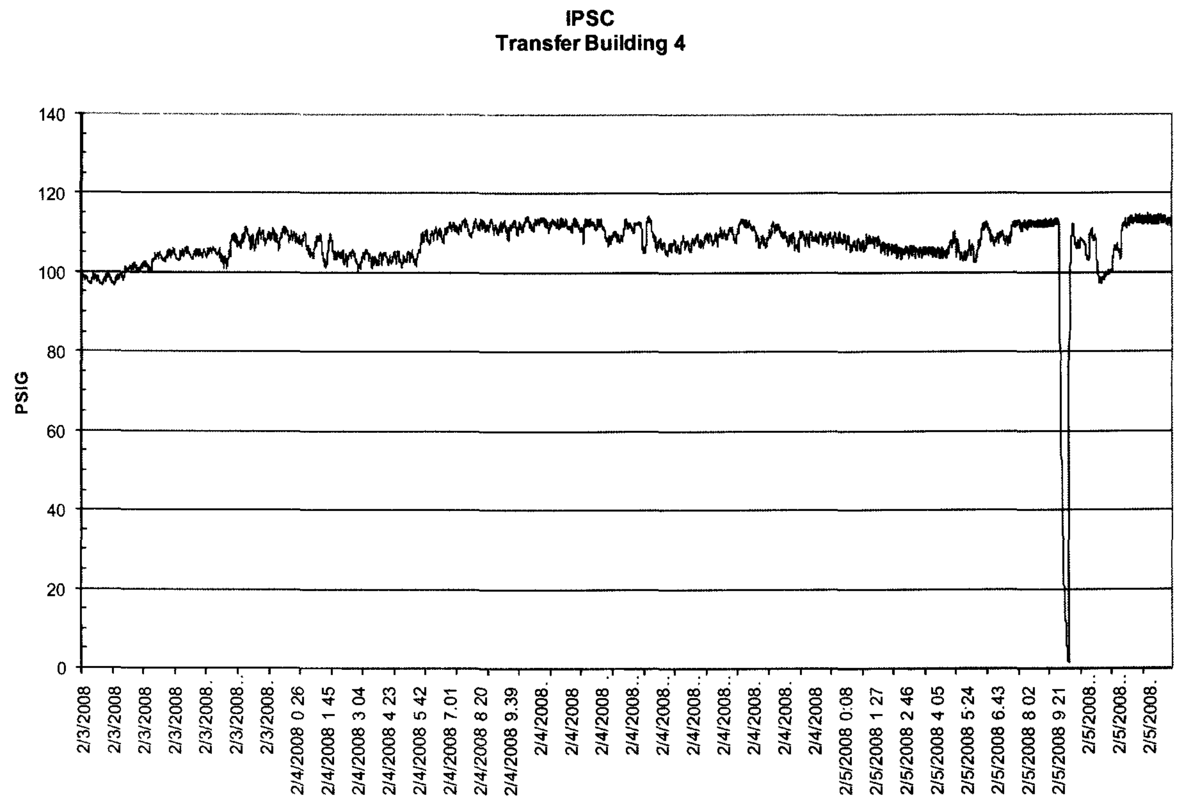

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Figure 26. Transfer Building #2

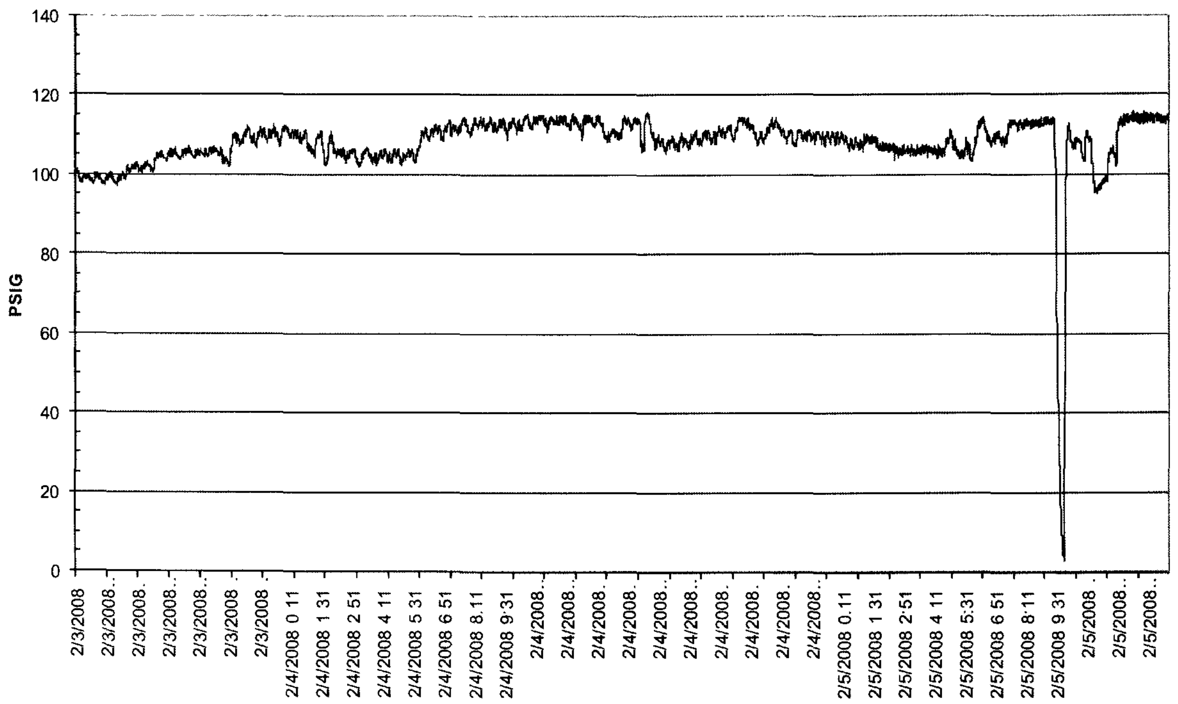

Figure 27. Transfer Building #1

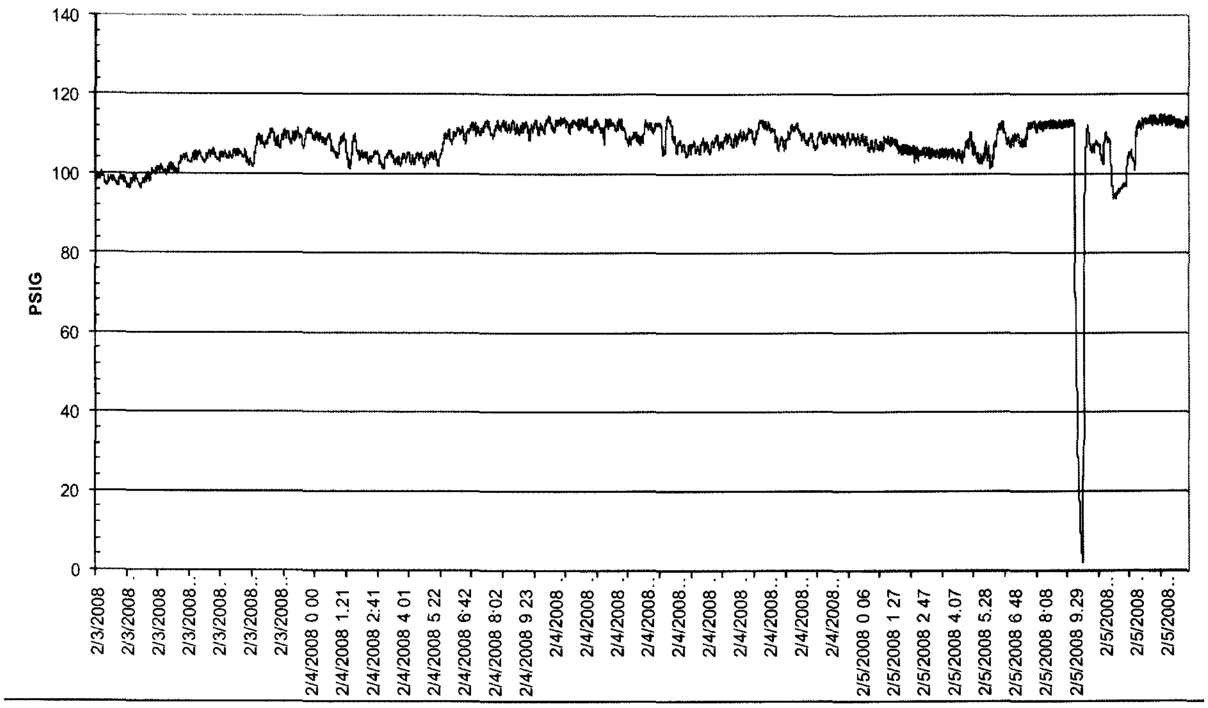

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Figure 28. Coal Car Unloading Receiver

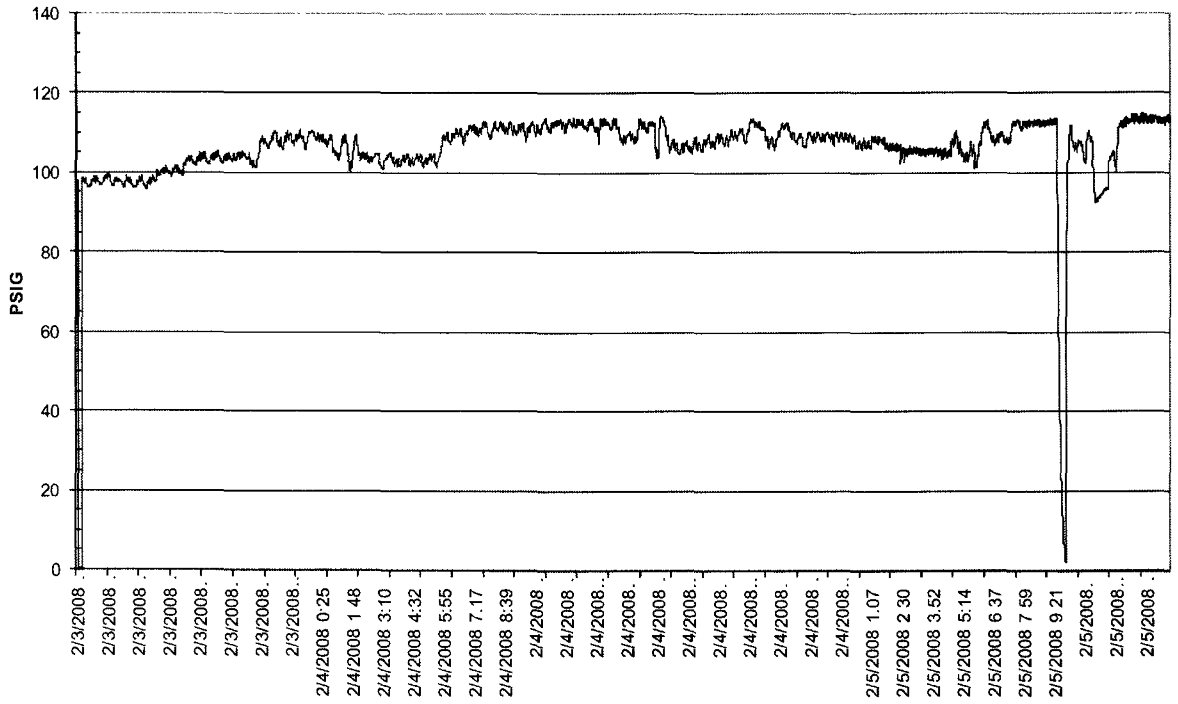

Figure 29. Limestone Unloading

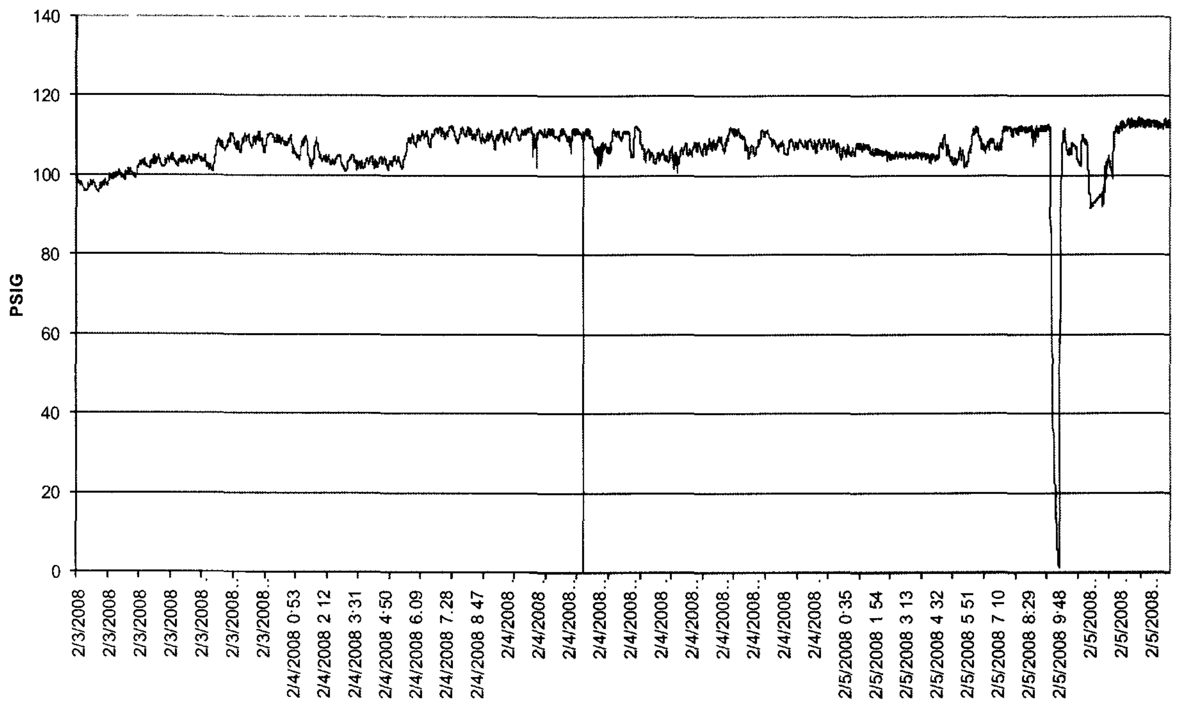

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Figure 30. Sludge Transfer

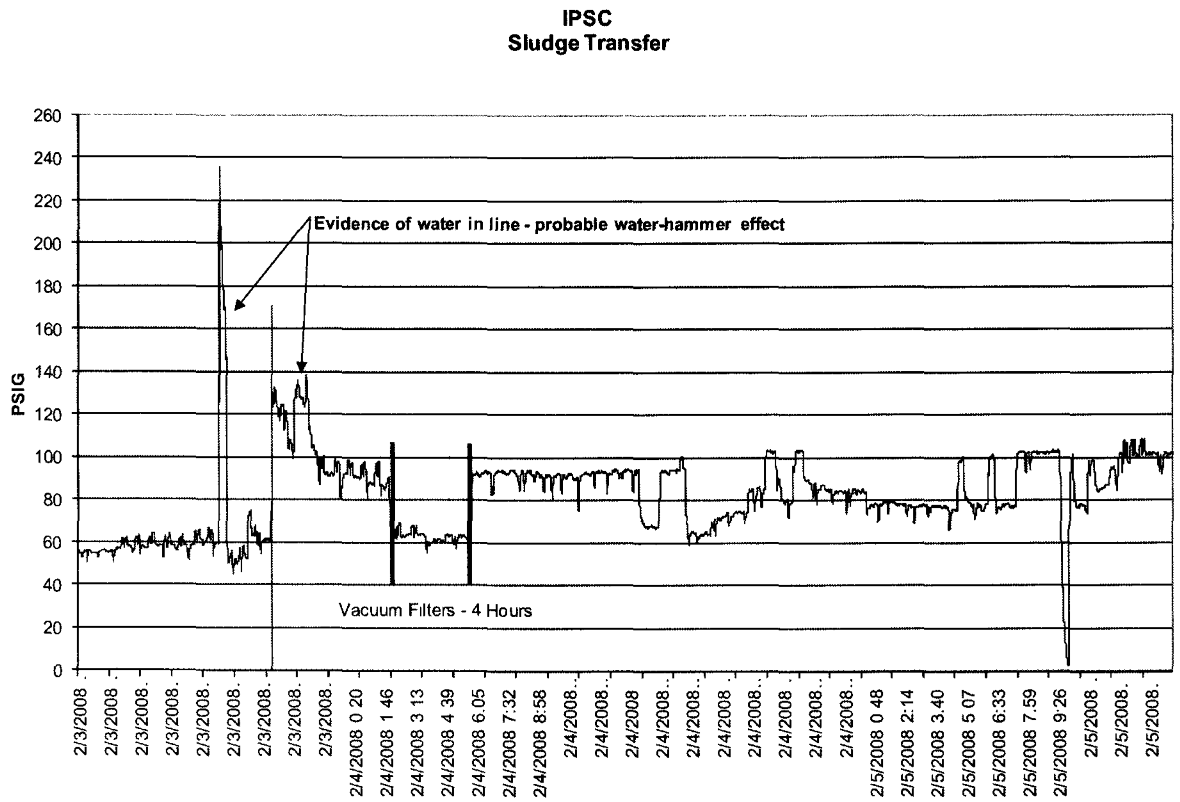

Figure 31. Stack

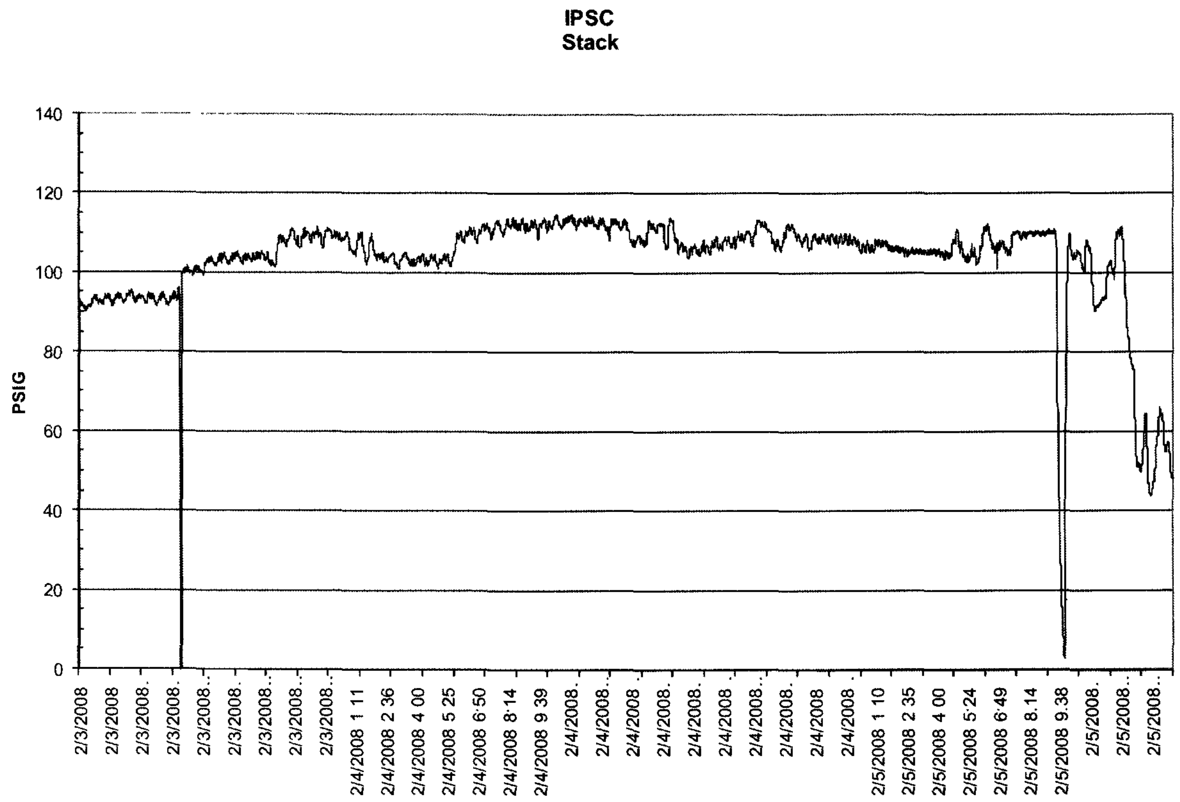

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Figure 32. Water Treatment

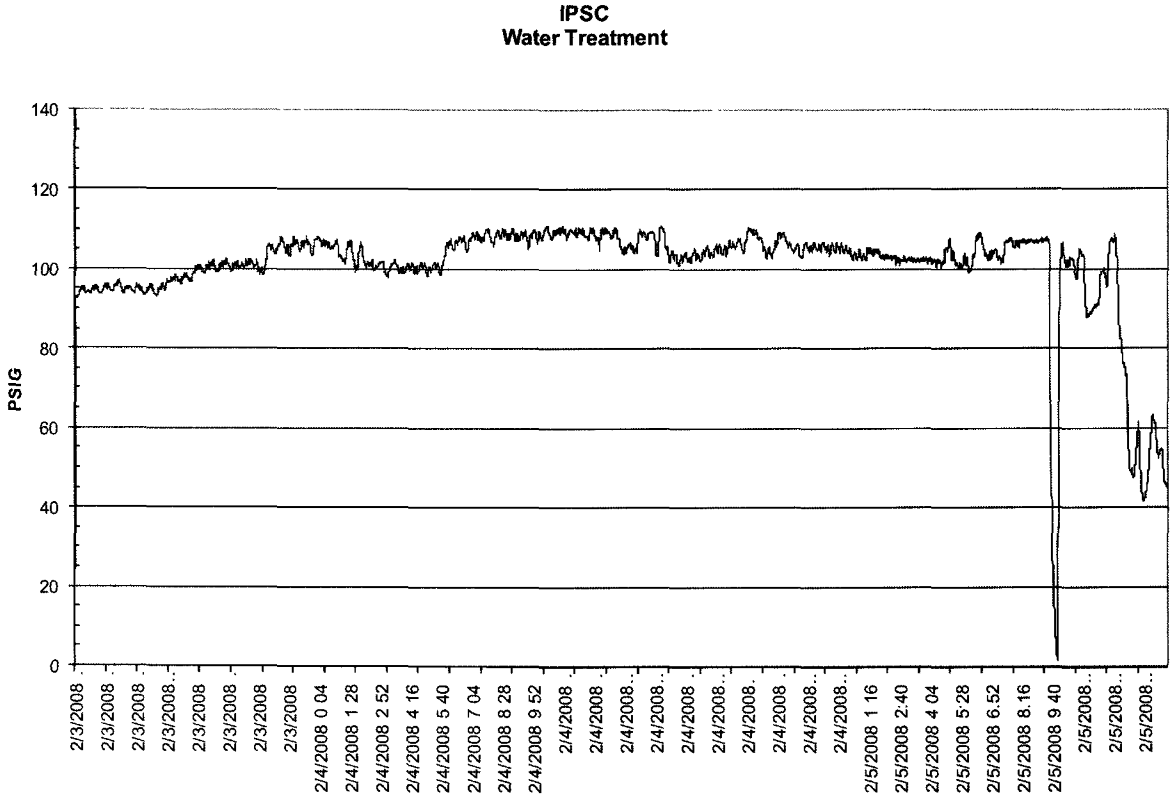


Figure 33. Boiler Sump Transfer

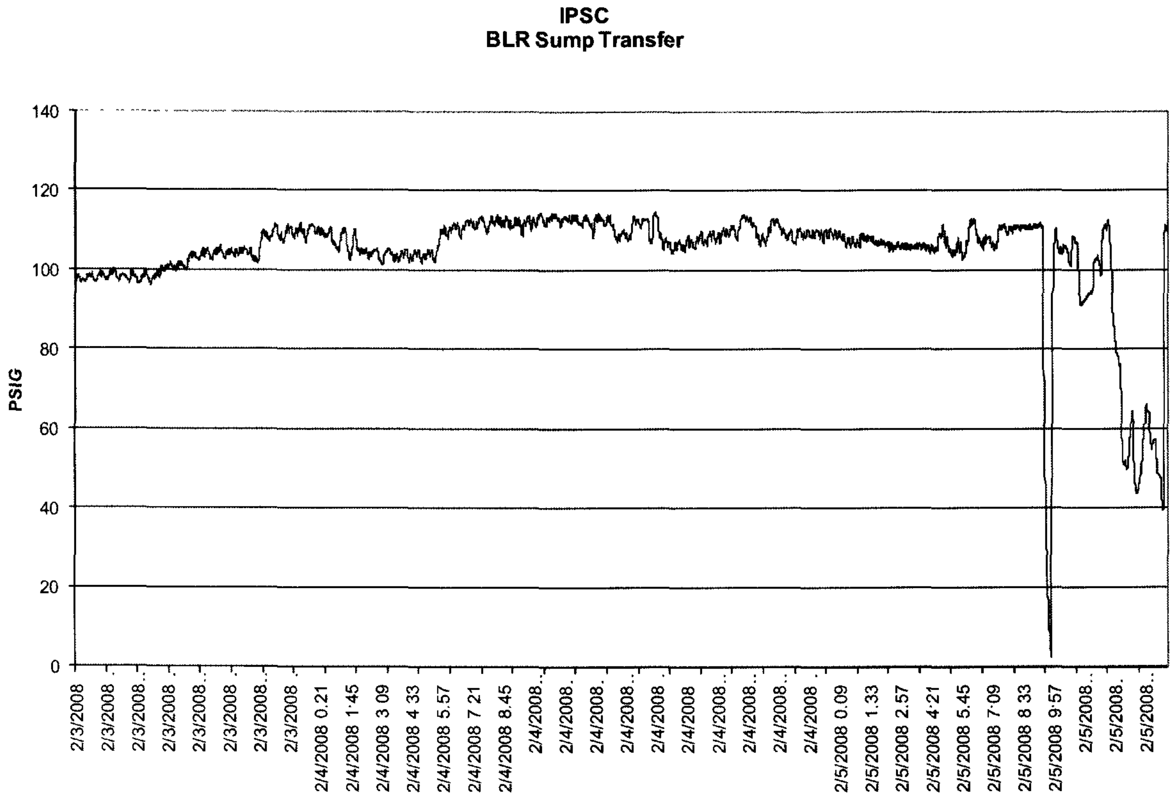


Figure 34. Igniter Boost Receiver

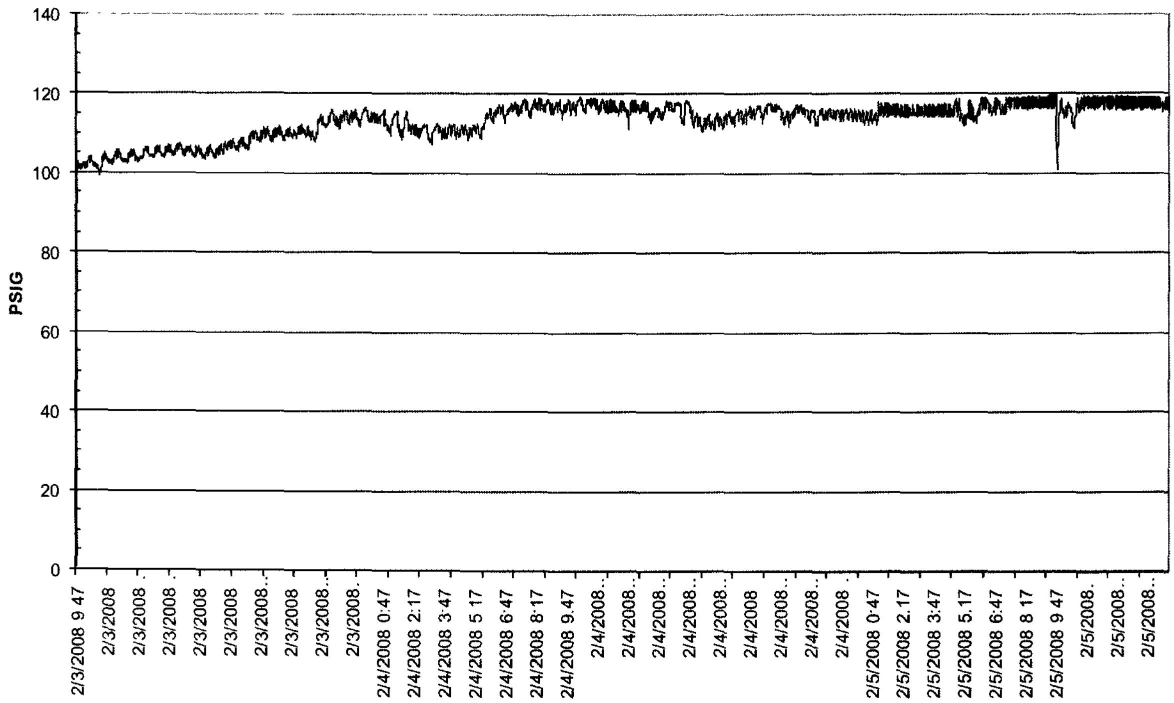
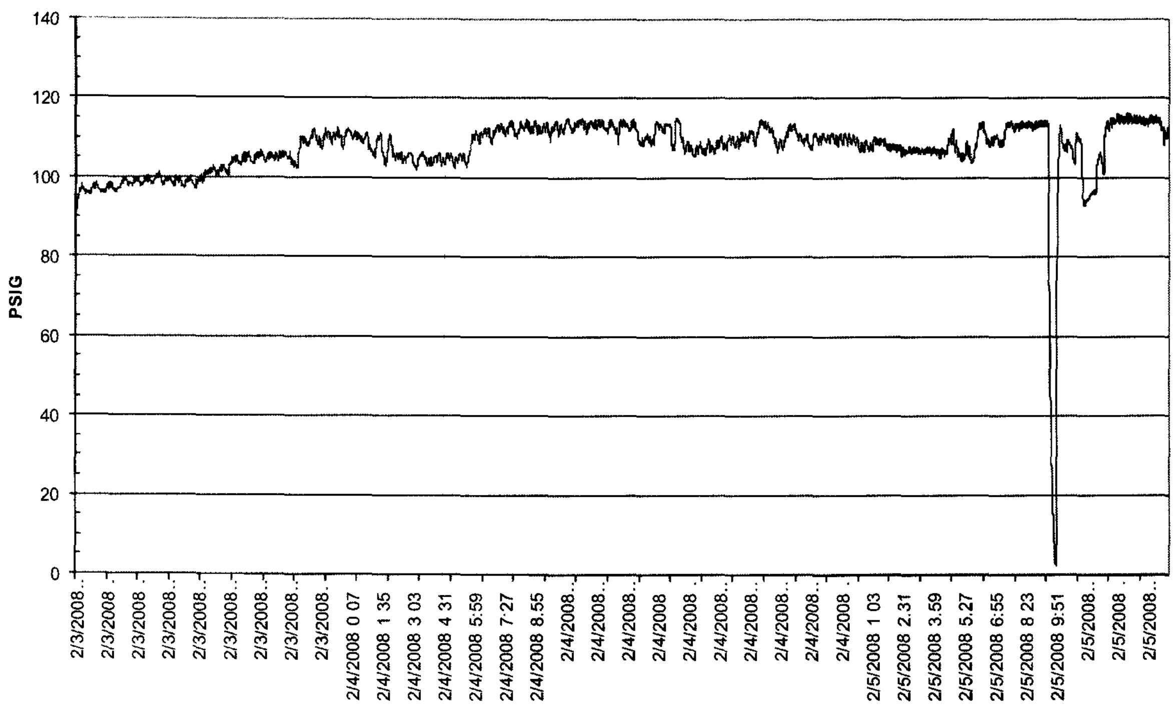


Figure 35. Lime Preparation Receiver



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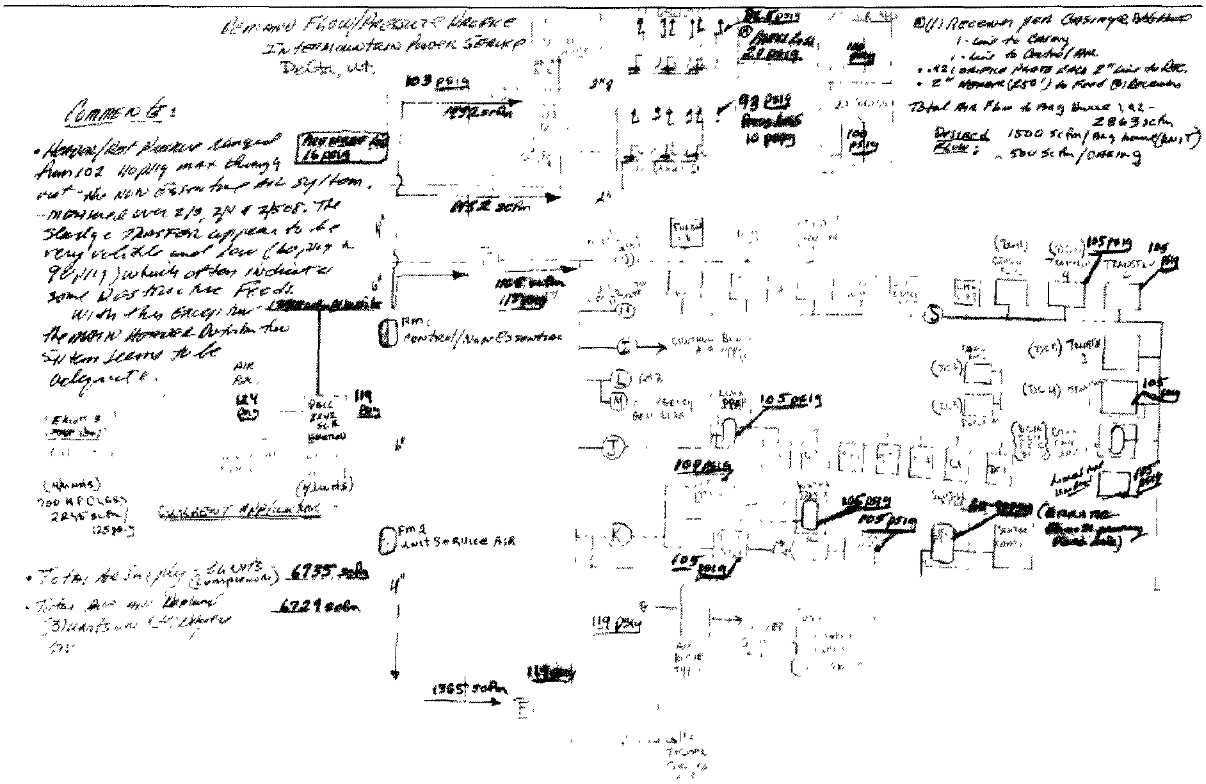

Play Layout foldout goes here

Figure 36. Plant Layout

4.2 PROCESS REGULATORS

Background

There are additional direct power cost savings to be obtained if the plant can continue to lower the overall system operating pressure. A steady delivered system pressure will allow follow-up programs at each process or point of use to establish the lowest overall effective pressure. This will enhance productivity, quality, and continue to reduce air usage and production costs.

The cornerstone of any effective demand-side air conservation program is to identify and operate at the lowest acceptable operating pressure required at all of the various production sectors and operating units in the plant. This should be a continuing program and part of any air system training or usage awareness program.

Some regulators are probably set at a higher-than-necessary feed pressure for an individual process, while others may be set wide open to full header pressure. Key questions to consider include: is there a minimum effective pressure established for each point of use for each production run? And, if so, is it being adhered to?

In this type of operation, it is very important that the actual inlet pressure for each process be known and that the lowest effective pressure be held steady for the proper product quality. Installation of storage bottles downstream of the regulator may be needed to "close up" the pressure readings at rest and at operation and offset regulator delay.

☑ **PHASE 2 RECOMMENDATION** – Review all regulated operations to establish lowest effective settings.

4.3 DUST COLLECTORS

Background

Proper operation of dust collectors is critical to minimizing cost and maximizing system effectiveness. There are many sizes and most, if not all, use a pulse of compressed air controlled by a timer. The timers are generally set by the operators to what they believe is appropriate for proper cake removal and bag life.

In a dust collection system, the dust is collected on the bag or fingers and when the cake of dust is of appropriate thickness and structure – a pulse or pulses of compressed air is used to hit or shock the bag and knock the cake off.

When the cake is removed correctly from the dust collector, the system removes dust from its assigned environment and has a normal bag life. When the cake is not removed effectively, the dust collector does not remove dust effectively from its assigned environment and the bag life can be significantly shortened.

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When the cake is removed correctly from the dust collector, the system removes dust from its assigned environment and has a normal bag life. When the cake is not removed effectively, the dust collector does not remove dust effectively from its assigned environment and the bag life can be significantly shortened.

Dust collection system designs specify the air inlet pressure to the manifold and pulse valves necessary for effective dust removal. The pulse valve sends a given volume or weight of air to the bag at a predetermined velocity to strike and clear the cake. The actual amount or weight of air is dependent upon the pulse nozzle being fed compressed air at a pre-determined and steady pressure.

The dust collector must receive the correct pressure (or close to it) and a steady repeatable pressure level for each pulse, particularly if timers are used to control the pulses. The operator may experiment to find the "right timing sequence" at a desired feed pressure. But if this pressure varies, then performance may not be satisfactory.

A problem that often occurs (short bag life) usually comes from the pulsers hitting the bag when the cake is not ready to flake off or the cake has gone too long between pulsing and grown too thick and heavy to clean effectively. This causes not only short bag life but very poor performance. There are usually several basic causes for this:

- Incorrect timer settings for the operating conditions. The actual requirement for the optimum timer setting may well change as various product runs change or even seasonally. These settings have to be set carefully to begin with and monitored regularly.

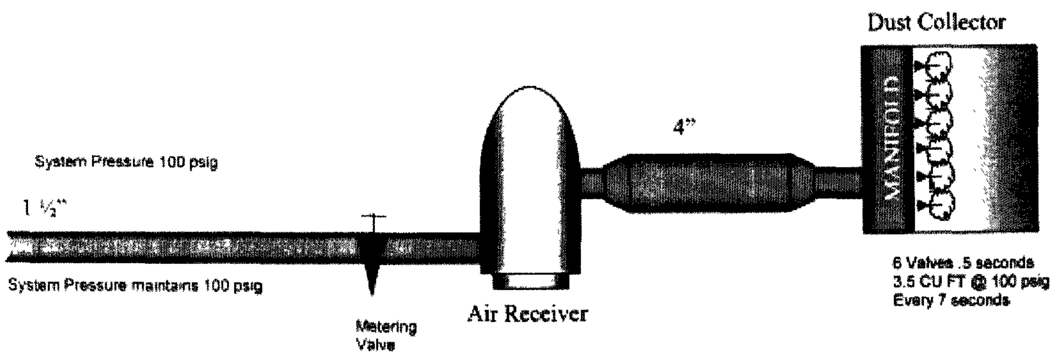
- Lack of sufficient storage or compressed air supply near the inlet manifold to supply the required pulse air without collapsing the inlet pressure. With too low an inlet pressure, the mass weight of the air pulse is too low, which then becomes ineffective in removing the cake.
- Too small a feed line to the dust collector will have the same effect as lack of air supply.
- Too small or incorrect regulator, which is unable to handle the required "Rate of Flow" required by the dust collectors.

All of these are installations or system situations that cause restricted air flow. They occur because, prior to the installation or prior to some operational change, the proper "rate of flow" was not identified for the dust collection action. Feed line sizing, regulator sizing, and air supply all require an identified "rate of flow." You cannot use "average flow rate."

"Flow rate" is the average flow or compressed air in cubic feet per minute either required by a process or delivered to the system. "Rate of flow" is the actual rate of flow of compressed air demand in cubic feet per minute. Even relatively small air demands in cubic feet can have a very high "rate of flow", if they occur over a very short time period. Dust collectors have this characteristic.

The sequence controllers can have a very significant impact on the required "rate of flow." For example, pictured here is a dust collector system, which has six pulsing valves that use 3.5 cu ft over 1/2 second for each pulse.

Problem: "Flow" / "Rate of Flow"



The impact of these two different "rates of flow" would show similar differences in regulator sizing, etc., if they are used on the feed line flow. The high flow velocities entering the manifold and controls for the pulse valves will create extra pressure loss through the balance affecting the performance of the pulse cleaner. The same sort of effect would show up in air receiver sizing to minimize system and feed line pressure drop if that is a question.

Typical Sizing (Each Valve Uses 3.5 scfm/pulse – 6 Valves on Collector)

Rate of Flow & Sizing with One Valve Hitting Every 7 Seconds	Rate of Flow & Sizing with Six Valves Hitting Every 7 Seconds
Rate of Flow = (1) x (3.5) = 3.5 x 60 ÷ .5 = 420 scfm	Rate of Flow = (6) x (3.5) 21 x 60 ÷ .5 = 2,520 scfm
The line size recommendation from the air supply to the dust collector = 90 psig line pressure = 2" to 3"	The line size recommendation from the air supply to the dust collector – 90 psig line pressure = 4" to 6"
• A 2" feed line will handle the 420 cfm flow at 90 psig line pressure with a velocity of 43 fps, which is about as high as it should go • A 3" feed line will handle the 420 cfm flow at 90 psig with a velocity of about 19 fps – very conservative • A 2" line would have a pressure loss of about 1 psid every 100' @ 420 scfm flow, which may be acceptable depending on feed line design and length • A 3" line would have pressure loss of less than .10 psid per 100' @ 420 scfm flow, which should be very acceptable	• A 5" feed line will handle the 2,520 cfm flow at 90 psig line pressure with a velocity of about 43 fps • A 6" feed line will handle the 2,520 cfm flow at 90 psig line pressure with a velocity of about 30 fps, which is conservative in this application • A 2" line at 2,520 cfm would have a minimum pressure loss of 30-50 psid, depending on timing and turbulence. This would be completely unacceptable • A 4" line would have a pressure loss of about 1.1 to 1.2 psid per 100' @ 90 psig and combined with moderate velocity should be acceptable depending on the length and design of the feed line • A 6" line would have a minimum pressure loss of .15 to .20 @ 90 psig with very low velocities and should be acceptable with "normal" installations

We recommend that every feed line has a quality pressure gauge installed near the dust collector entry. Observe the pressure gauge, which the pulser hits – if the pressure drop is too high (over 10-20 psig), start looking for the cause. Get the specification on the dust collector, cfm per pulse, feed line pressure time per pulse, cycle time between pulses, etc. Calculate the rate of flow, check line size and storage. If additional storage is required, this can be calculated by the following formula:

For example, 2,520 rate of flow @ .5 seconds flow with 4 psig allowable pressure loss.

Size Air Receiver: $T_{min} = \dfrac{(V)\,(P_2 - P_1)}{(CFM)\,(14.5)}$

Net Rate of Flow 2,520 cfm

P_1 – Rest Pressure 100 psig

P_2 – Allowable Drop 96 psig (4 psig)

$T_{sec} = \dfrac{(V)\,(4)\,(60)}{.5 \text{ sec } (2{,}520)(14.4)} = .5 \text{ sec} = \dfrac{240\text{ V}}{36{,}540}$

240 V = 18,270 V = 76 cu ft x 7.48 = 570 gal or more

Refill Rate of Flow:

Time allowed – 6 seconds

21 cu ft x 60 seconds ÷ 6 seconds =

210 cfm rate of flow

Effect on Header: Negligible

We have used storage to convert a high rate of flow to a low rate of flow and eliminate system pressure collapse

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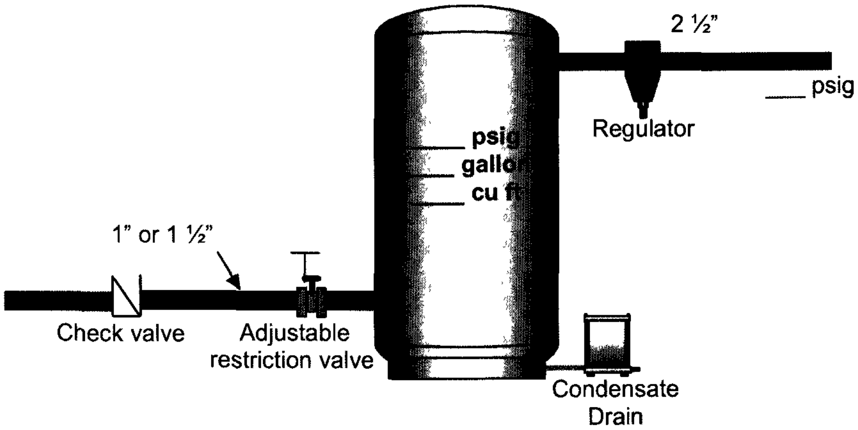
Significant amounts of air (10 to 15 cfm or more) can be lost when the control diaphragm and/or connections fail. Such leaks are very difficult to find and repair.

Proper sizing and installation of appropriate storage for dust collectors offers opportunities to convert high volume short term demand to lower average rate of flow.

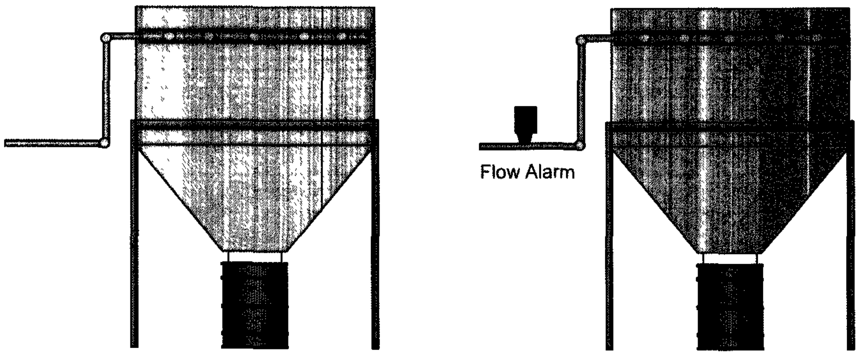
Install an appropriate sized receiver near the process (dust collector, etc.). System air at 90 psig* from inlet and regulate to each feed line to dust collector. Size the regulator to handle appropriate scfm rate of flow with minimum inlet pressure of 92 psig. Install check valve and adjustable restriction valve on inlet line to air receiver.

(* Pressure must exceed the minimum for process.)
For example, Pulse is 3.5 cfm ½ sec every 7 seconds (see preceding page)



Adding appropriate storage may not only be a direct energy issue but one of air quality. Proper control of the dust collectors will protect surrounding systems from falling pressure at nozzle blow. This should also enhance the dust collector performance and extend bag life.

Dust collectors are a significant source of leaks that are hard to detect. Often the pulse control diaphragms leak. An electronic air flow alarm can signal this problem visually and remotely.



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Current Application

Some of the dust collector feeds appear to be marginally sized, and each does not have an air receiver between it and the collector. Observation of the operation and discussions with plant personnel indicate the demand controls are working well and the bags sloughing off properly. There does not appear to be a problem of pulling low pressure in surrounding lines but their does appear to be some problem pulling the inlet pressure down to the collector drain at the pulse (see the following data collected at the Limestone unload).

The procedures shown on the preceding pages indicate how to size a proper volume receiver to avoid this.

The Limestone unloading dust collectors may operate 4-6 hours a day and the conveyors 14-15 hours per day. However, our observation indicates:

Due to the successful DP control system, the actual cleaning time is much less.
The pulses appear to be well controlled.

Summary

There does not appear to be a significant air saving opportunity here at this time. However, there does appear to be some action that can be taken to stabilize the actual feed pressure to the pulses and avoid today's turndown.

Observed Operation of Limestone Unloading Dust Collector

Date:	6 February 2008
Time:	10:30 – 11:30 am
Inlet:	Press to dust collector 1" feed line; 35' long – regulator and filter – set to 90 psig

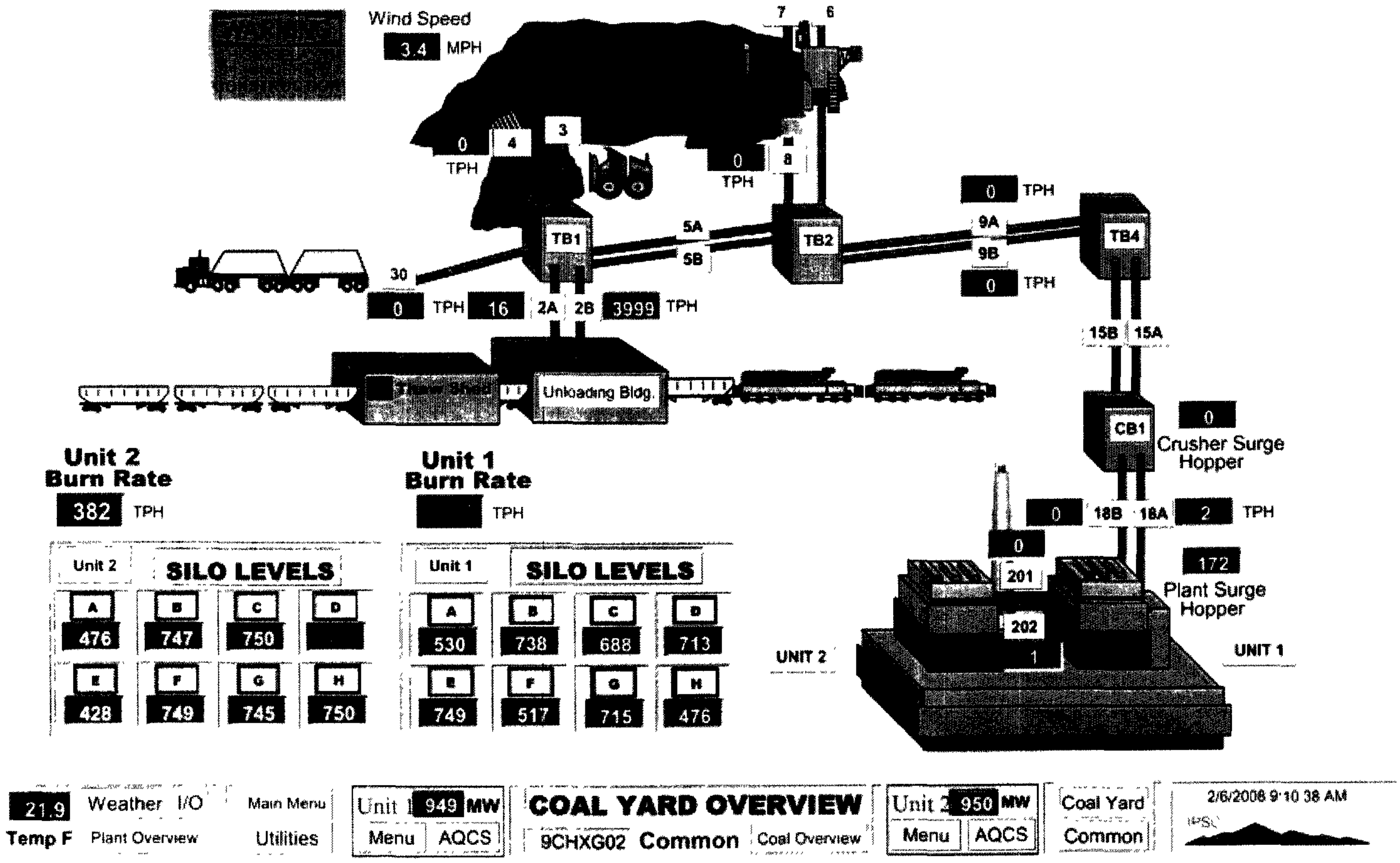
The plant was advised to run at 80 psig pulse pressure to avoid bag premature deterioration wit the Teflon-coated bag. The regulator was set higher to avoid low pressure shut off.

Inlet pressure fell at pulse (6 pulsers):

	Drop		Actual Inlet
#1	30 psig	to	60 psig
#2	30 psig	to	60 psig
#3	38 psi	to	52 psig (possible "blown bag")
#4	39 psig	to	51 psig (possible "blown bag")
#5	30 psig	to	60 psig
#6	30 psig	to	60 psig

The above data shows that actual inlet pressure to the pulser is not the recommended 80 psig, but 60 psig. This may be having a negative impact on bag filter integrity. We recommend investigation when convenient to see if this is or is not a problem to be addressed. If it is a problem, the methodology for correction is in the preceding sample or we will be happy to work with you on this.

Figure 37. Coal Yard Overview



4.4 LEAK IDENTIFICATION AND REPAIR

Most plants can benefit from an ongoing leak management program. Generally speaking, the most effective programs are those that involve the production supervisors and operators working in concert with the maintenance personnel. Accordingly, it is suggested that all programs consist of the following:

- **Short Term** – Set up a continuing leak inspection by Maintenance Personnel so that for a while, each primary sector of the plant is inspected once each quarter to identify and repair leaks. A record should be kept of all findings, corrective measures, and overall results. The PROJECT COST SECTION below includes current price quotes for ultrasonic leak locator equipment.

- **Long Term** – Consider setting up programs to motivate the operators and supervisors to identify and repair leaks. One method that has worked well with many operations is to monitor the air flow to each department and make each department responsible for identifying its air usage as a measurable part of the operating expense for that area. This usually works best when combined with an effective in-house training, awareness, and incentive program.

With a plant of this type, an effective leak management program could save 1,500 cfm or the equivalent of repairing 500 leaks averaging 3 cfm each. On a percentage basis, this leak level is about the same as leak levels in other plants. Repairing leaks totaling 1,500 cfm translate into an annual savings of $152,505 per year.

Compressed Air Leak Survey

A survey of compressed air leaks was conducted at the plant and 174 leaks were identified, quantified, tagged, and logged. Potential savings totaled 855 cfm for the 174 leaks that were identified.

We recommend an ultrasonic leak locator be used to identify and quantify the compressed air leaks. We use either a VXP AccuTrak manufactured by Superior Signal or a UE Systems Ultraprobe 2000.

Shutting off the air supply to these leaks when the area is idle would save significant energy use. Reducing the overall system pressure would also reduce the impact of the leaks, when air to the machine cannot be shut off. Repairing the leaks can save additional energy. The savings estimates associated with a leak management program are based on the unloading controls of the compressors being able to effectively translate less air flow into lower cost.

With a few minor exceptions, most of the leaks could not have been found without the use of an ultrasonic leak detector and a trained operator. Leak locating during production time with the proper equipment is very effective and often shows leaks that are not there when idle. However, a regular program of inspecting the systems in “off hours” with “air powered up” is also a good idea. In a system such as this one, some 90 to 95% of the total leaks will be in the use of the machinery, not in the distribution system.

The area surveyed in the leak study included a great deal of high background ultrasound noise that shields many of the smaller leaks. In continuing the leak management program, plant staff

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should perform leak detection during non-production hours in order to eliminate some of the high ultrasonic background noise.

☑ **RECOMMENDED PROJECT (#7)** – Implement ongoing leak identification and repair program with ultrasonic locators. Repair all tagged leaks listed on the following page.

Estimated reduction of air flow with proposed project	855 cfm
Recoverable savings from air flow reduction [Section 2.3]	$84.53 /cfm yr
Annual electric cost savings with proposed project	$72,273 /year
Cost of leak detection equipment (if required)	$2,800
Number of leaks	178
Estimated cost of leak repairs ($100 per leak)	$17,800
Total project cost (materials and installation)	$20,600

☑ **PHASE 2 RECOMMENDATION** – Continue an aggressive leak tagging and repair program. Quantify and value the leak(s) and report to management on a predetermined regular basis.

Leak List

TAG	LOCATION	DESCRIPTION	EST. SIZE	EST. CFM.
3829	COMPRESSOR ROOM	COMPRESSOR 1A A/C BLOWDN & VLV	LARGE	20
3829	COMPRESSOR ROOM	1D COMPRESSOR A/C DRAIN	VERY LG	70
3830	UNIT 2 BAGHOUSE	A CASING CYL 1087	LARGE	15
3831	UNIT 2 BAGHOUSE	A CASING CYL 1103	SMALL	3
3832	UNIT 2 BAGHOUSE	A CASING CYL 1104	SMALL	3
3833	UNIT 2 BAGHOUSE	A CASING PURGE AIR	SMALL	3
3834	UNIT 2 BAGHOUSE	CYL LEAK--C16 CYLINDER 3147	SMALL	4
3835	UNIT 2 BAGHOUSE	CYL LEAK--C13 CYLINDER 3120	SMALL	2
3836	UNIT 2 BAGHOUSE	CYL LEAK--C12 CYLINDER 3110	SMALL	3
3837	UNIT 2 BAGHOUSE	CYL LEAK--C11 CYLINDER 3101	SMALL	3
3838	UNIT 2 BAGHOUSE	CYL LEAK--C3 CYLINDER 3030	SMALL	2
3839	UNIT 2 BAGHOUSE	CYL LEAK--C1 CYLINDER 3012	SMALL	2
3840	UNIT 2 BAGHOUSE	CYL LEAK--C9 CYLINDER 3083	SMALL	4
3841	UNIT 2 BAGHOUSE	CYL LEAK--B1 CYLINDER 2012	SMALL	2
3842	UNIT 2 BAGHOUSE	CYL LEAK--B1 CYLINDER 2011	SMALL	2
3843	UNIT 2 BAGHOUSE	CYL LEAK--B11 CYLINDER 2101	SMALL	3
3844	UNIT 2 BAGHOUSE	CYL LEAK--B11 CYLINDER 2102	SMALL	2
3845	UNIT 2 BAGHOUSE	CYL LEAK--B3 CYLINDER 2030	SMALL	3
3846	UNIT 2 BAGHOUSE	CYL LEAK--B12 CYLINDER 2110	SMALL	2

3847	UNIT 2 BAGHOUSE	CYL LEAK--B12 CYLINDER 2111	SMALL	3
3848	UNIT 2 BAGHOUSE	CYL LEAK--B4 CYLINDER 2038	SMALL	3
3849	UNIT 2 BAGHOUSE	CYL LEAK--B13 CYLINDER 2119	SMALL	4
3850	UNIT 2 BAGHOUSE	CYL LEAK--B13 CYLINDER 2120	SMALL	3
3851	UNIT 2 BAGHOUSE	CYL LEAK--B14 CYLINDER 2128	SMALL	2
3852	UNIT 2 BAGHOUSE	CYL LEAK--B14 CYLINDER 2129	SMALL	2
3853	UNIT 2 BAGHOUSE	CYL LEAK--B7 CYLINDER 2065	SMALL	2
3854	UNIT 2 BAGHOUSE	CYL LEAK--B15 CYLINDER 2133	SMALL	4
3855	UNIT 2 BAGHOUSE	CYL LEAK--B8 CYLINDER 2075	SMALL	2
3856	UNIT 2 BAGHOUSE	CYL LEAK--A8 CYLINDER 1075	SMALL	2
3857	UNIT 2 BAGHOUSE	CYL LEAK--A15 CYLINDER 1138	MED	5
3858	UNIT 2 BAGHOUSE	CYL LEAK--A14 CYLINDER 1129	SMALL	3
3859	UNIT 2 BAGHOUSE	CYL LEAK--A13 CYLINDER 1120	SMALL	3
3860	UNIT 2 BAGHOUSE	CYL LEAK--A13 CYLINDER 1119	SMALL	2
3861	UNIT 2 BAGHOUSE	CYL LEAK--A4 CYLINDER 1038	SMALL	2
3862	UNIT 2 BAGHOUSE	CYL LEAK--A10 CYLINDER 1093	SMALL	3
3863	UNIT 2 BAGHOUSE	CYL LEAK--A2 CYLINDER 1020	SMALL	3
3864	UNIT 2 BAGHOUSE	1A01 CYL 1011	SMALL	3
3865	UNIT 2 BAGHOUSE	C-CASING CYL 3085--SOL EXH	SMALL	2
3866	UNIT 2 BAGHOUSE	C-CASING CYL 3094--TOP	MED	5
3867	UNIT 2 BAGHOUSE	C-CASING CYL 3114	SMALL	3
3868	UNIT 2 BAGHOUSE	C-CASING CYL 3151	SMALL	3
3869	UNIT 2 BAGHOUSE	C-CASING CYL 3076	SMALL	3
3870	UNIT 2 BAGHOUSE	C-CASING CYL 3070	SMALL	3
3871	UNIT 2 BAGHOUSE	C-CASING CYL 1007 PURGE	SMALL	3
3872	UNIT 2 BAGHOUSE	C-CASING CYL 3033	MED	6
3873	UNIT 2 BAGHOUSE	C-CASING CYL 3022	SMALL	2
3874	UNIT 2 BAGHOUSE	C-CASING REG--AIR HORN AIR	SMALL	2
3875	UNIT 2 BAGHOUSE	C CASING CYL 1C01 TOP	SMALL	3
3876	UNIT 2 BAGHOUSE	B CASING CYL 2150	SMALL	2
3877	UNIT 2 BAGHOUSE	B CASING CYL 2140	SMALL	2
3878	UNIT 2 BAGHOUSE	B CASING 3/4" FILTER	MED	5
3879	UNIT 2 BAGHOUSE	B CASING CYL 2122	SMALL	3
3880	UNIT 2 BAGHOUSE	B CASING CYL 2121	SMALL	3
3881	UNIT 2 BAGHOUSE	B CASING CYL 2115	SMALL	3
3882	UNIT 2 BAGHOUSE	B CASING CYL 2113	SMALL	3
3883	UNIT 2 BAGHOUSE	B CASING CYL 2112	SMALL	2
3884	UNIT 2 BAGHOUSE	B CASING CYL 2104	SMALL	2
3885	UNIT 2 BAGHOUSE	B CASING CYL 2096	SMALL	2
3886	UNIT 2 BAGHOUSE	B CASING CYL REV AIR	MED	5
3887	UNIT 2 BAGHOUSE	B CASING CYL 2014	SMALL	3
3888	UNIT 2 BAGHOUSE	B CASING CYL 2040	MED	6
3889	UNIT 2 BAGHOUSE	B CASING CYL 2041	SMALL	3
3890	UNIT 2 BAGHOUSE	B CASING CYL 2059A	SMALL	3
3891	UNIT 2 BAGHOUSE	B CASING CYL 2068	SMALL	3
3892	UNIT 2 BAGHOUSE	A CASING CYL 1014	SMALL	3
3893	UNIT 2 BAGHOUSE	A CASING CYL 1032--SOL VLV	MED	5
3894	UNIT 2 BAGHOUSE	A CASING CYL 1050	SMALL	3

3895	UNIT 2 BAGHOUSE	A CASING CYL 1068	LARGE	10
3896	UNIT 2 BAGHOUSE	A CASING CYL 1076	SMALL	2
3897	UNIT 2 BAGHOUSE	A CASING CYL 1130	MED	5
3898	UNIT 2 BAGHOUSE	A CASING CYL 1115	MED	5
3899	UNIT 2 BAGHOUSE	A CASING CYL 1114	SMALL	3
3899	UNIT 2 BAGHOUSE	A CASING CYL 1105	SMALL	3
	Unit 1 Casing Bag House			
3900	1A stnd 35	ZSL block vent/lower	small	3
3901	1A bypas #3	mositure trap/site glass	small	3
3902	1A bypass north wall	mositure trap/bottom fitting	small	3
3903	Reverse Air Relief damper	cyld packing/when up position	small	3
3904	stand 1 cyld	packing seal/when up position	small	4
3905	stand 2 cyld	packing seal/when up position	small	4
3906	stand 3	muffler filter/upper muffler	medium	5
3907	stand 9 cyld	packing seal/when up position	small	3
3908	stand 21	muffler filter/lower muffler	small	3
3909	stand 22	muffler filter/upper muffler	small	3
3910	stand 23 cyld	packing seal/when up position	small	4
3911	stand 32	ZSL block vent/upper vent	medium	5
3912	stand 34 cyld	packing seal/when up position	small	3
3913	stand 35 cyld	packing seal/when up position	medium	5
3914	stand 42	muffler filter/upper muffler	medium	5
	stand 42 cyld	packing seal/when up position	large	10
3915	stand 51 cyld	packing seal/when up position	small	3
3916	stand 55 cyld	packing seal/when up position	medium	4
3917	stand 57	ZSL block vent/upper vent	small	3
3918	stand 58	muffler filter/upper muffler	small	3
3919	stand 59	muffler filter/upper muffler	small	3
3920	FF compt 1A6	ZSH block vent/left cyld	small	4
3921	FF compt 1B14	ZSH block vent/left cyld	small	3
3922	FF compt 1B5	ZSH block vent/left cyld	medium	5
3923	FF compt 1B4	ZSH block vent/left cyld	medium	6
3924	FF compt 1B12	ZSH block vent/right cyld	small	3
3925	FF compt 1B11	ZSH block vent/right cyld	medium	4
3926	FF compt 1B10	ZSH block vent/right cyld	small	3
3927	FF compt 1C5	ZSH block vent/right cyld	small	3
3928	B Casing relief cyld	packing seal/when up position	medium	4
3929	stand 1 cyld	ZSL block vent/lower block vent	small	3
3930	stand 2	ZSL block vent/upper vent	small	3
3931	stand 6	ZSL block vent/upper vent	small	3
3932	stand 9	muffler filter/lower muffler	small	4
3933	STAND 9	ZSL block vent/lower vent	small	3
3934	stand 10	muffler filter/upper muffler	medium	5
3935	stand 11 cyld	packing seal/when up position	small	3
3936	Between stand 12 & 13	Lubricator/top piece	medium	8
3937	stand 14 cyld	packing seal/when up position	medium	6
3938	stand 19	electric solenoid/attaches to air block	small	2

3939	stand 20	muffler filter/lower muffler	small	3
3940	purge Hdr between 36/37	moisture trap/bottom of bowl	medium	8
3941	stand 34	ZSL block vent/lower block vent	small	2
3942	stand 38	ZSL block vent/lower block vent	medium	4
3943	stand 41	ZSL block vent/upper block vent	small	3
3944	stand 45	muffler filter/lower muffler	small	3
3945	stand 49	muffler filter/lower muffler	small	3
3946	stand 50	muffler filter/upper muffler	small	2
3947	stand 50	ZSL block vent/lower block vent	small	3
3948	stand 54	muffler filter/upper muffler	small	3
3950	stand 54	ZSL block vent/upper block vent	small	2
3951	stand 60	ZSL block vent/lower block vent	small	2
3952	stand 62	muffler filter/upper muffler	medium	6
3953	stand 64	ZSL block vent/upper block vent	small	2
3954	Casing C Relief damper	cyld packing/when up position	medium	5
3955	stand 8	ZSL block vent/upper block vent	small	2
3956	Stand 12	ZSL block vent/upper block vent	small	2
3957	stand 16	ZSL block vent/lower block vent	small	2
3958	stand 31 cyld	packing seal/when up position	small	3
3959	stand 31	muffler filter/upper muffler	small	3
3960	stand 38	muffler filter/upper muffler	large	15
3961	stand 44	ZSL block vent/upper block vent	small	2
3962	stand 48	muffler filter/lower muffler	small	3
3963	stand 55 cyld	muffler filter/upper muffler	medium	5
3964	bag house FA feeder 1C9	*solenoid valve body/*	small	2
3965	FA feeder 1C10	solenoid valve body/	small	3
3966	FA feeder 1C13	solenoid valve body/	small	4
3967	FA feeder 1C16	air cyld/over head	small	2
3968	Fa feeder 1C5	solenoid valve body/	small	3
3969	FA feeder 1B15	solenoid valve body/over head with rag on it	large	15
3970	FA feeder 1A12	air cyld/overhead	small	2
3971	FA feeder 1A14	air cyld/overhead	small	3
3972	FA feeder 1A15	air cyld/overhead	small	3
3973	conveyor 18A-B	Air gear box /(2) ball valve is venting	large	15
3974	Conveyor 18 A-B	air brake/both ball valves venting 100% of time	Large	15
3975	Conveyor 9	#1 air drop leg/Bad Ball valve leaking when on	Medium	5
3976	Conveyor 5	#2 air drop leg/Bad Ball valve leaking when on	Medium	5
3977	Conveyor 5	#3 air drop leg/Bad Ball valve leaking when on	Medium	4
3978	Transfer Building 1	Dust collector/Air valve accuator on top of silo	Large	15
3979	B train water treatment	9WTD-ABV-227/middle of valve body	Small	3
3980	A train water treatment	9WTD-ANX-3A/regulator	Small	3
3981	A train water treatment	9WTD-ABV-96/middle of valve body	Small	3
3982	A train water treatment	9WTD-ABV-25/Diaphgram	Small	3
	Paint Shop	moisture trap/cracked bowl	Small	4
No Tag	Dust collector B	Damper valve on silo/Air valve accuator on top of silo	Large	25
No Tag	Dust collector D	Damper valve on silo/Air valve accuator on top of silo	Large	25

A1	Sludge C	Thickner Tunnel/under stairs soleniod blowing	Large	15
A2	Sludge Tank Bldg	ABV 221/up stairs hose	Medium	4
A3	Sludge bld 4	ABV 6/air line fitting	Small	3
A4	lime prep	bad ABV/blowing air block valve	Large	10
A5	Scrubber 2 2nd flr	ABV 545/E module fitting	Medium	4
A6	Scrubber 2 2nd flr	ABV 546/F module fitting	Medium	4
A7	Scrubber 2 2nd flr	ABV/A module blowingAir BV	Large	15
A8	Scrubber 2 1st flr	W sump pp disch valve/Leaking from cyld green ABV	Large	15
A9	Scrubber2 2	pp dich line flush ABV/3D ABV bad hose fitting	Medium	5
A10	Scrubber 1 4th flr	Air BV/by elevator	Medium	5
A11	Scrubber 1 3rd flr	Quench vlv cyld/F module	Medium	5
A12	Scrubber 1 4th flr	Bad ABV/NE stair bad ABV	Large	10
A13	Scrubber 1 C module	Mist Elim ABV 195/2 CCC air leak around diaphram	Large	10
A14	Scrubber 1 B Module	Temp. pacth leak/2nd flr ground level	Large	10
A15	Scrubber 1 1st flr	Cracked pipe ABV/2F SP PP F module	Large	10
			Total:	**855**

4.5 POTENTIALLY INAPPROPRIATE USES OF COMPRESSED AIR

Potentially inappropriate uses of compressed air are demand-side applications that may be more efficiently handled by another power source rather than compressed air. The following sections identify and evaluate some inappropriate applications present at many plants.

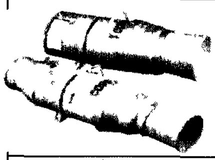
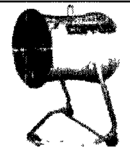
4.5.1 Air Movers or Air Horns

These items are part of a family of products known as "Portable Ventilators." They are available in various designs to move large volumes of air (1,000 to 10,000 cfm) in plants for many applications. The most common drives are electric, but they also come in Venturi air drives, which use high pressure compressed air to pull ambient outside air by a Venturi action. Generally, these use from 100 scfm each to 300 scfm for the most common 6" and 8" sizes.

The extensive use of air horns throughout the plant during the hot summer months (7 months). Can create a significant continuing load demand not accounted for in this audit.

The chart below lists the most popular 6" and 8" air horns and their annual electrical energy operating cost.

Operating Cost Comparison for Air Horns

	Model / Class	Air Flow to Process	Compressed Air Usage @ 60 psig	Energy Cost	Electric HP	Energy Cost	Estimated Price per Unit
	TX6AM Compressed Air	2,885 cfm	98 (24 kW)	**$6,000/yr**			$400 - $600
	TX8AM Compressed Air	4,152 cfm	152 (41.4 kW)	**$10,350/yr**			$400 - $600
	VANO 250 Electric	3,000 cfm			1 HP	**$200/yr**	$2,000
	Double Heat Killer Electric	9,500 cfm			1 HP	**$200/yr**	$3,000
	TA 16 Electric	5,500 cfm			2 HP	**$400/yr**	$2,600

Operating cost based on $0.05 /kWh – 5,000 hours/year.

- The compressed air-driven horns have a significant lower initial cost.
- The electrical energy cost savings of the electric-driven alternatives creates a very quick, simple payback.
- The air flows shown are at 80 psig; at 100 psig, they would be 20% higher.
- Vortec fixed flow vortex coolers are also often used to spot cool bearings. The approximate performance characteristics are:

Standard Model			
Size	1"	2"	3"
Compressed Air @ 80 psig	50 scfm	75 scfm	100 scfm
Temperature Drop / 80 psig	60-80°F	60-80°F	60-80°F

The most popular size is the 100 cfm cooler. As long as these are used for spot cooling at or on a bearing, etc., for a limited time, they may be the best choice for effectiveness. If these can be replaced with a Vano electric-operated coolers, the savings would be about 100 cfm each (16.6 kW or $4,050 per year in basic electric cost. These are significantly more expensive or less efficient than the standard air horns. We recommend that these be changed as fast as possible.

Reviewing the electric-driven air movers available from Coppus (and others), the following units would be appropriate substitutes for air horns to be used for cooling and ventilation and offer significantly savings.

- First choice for directed air flow such as cooling:

  VANO 175 CV or 250 CV – these are powered by electric motor-driven axial vane fans, capable of large volume flow through ducting as required. They are available with totally enclosed motors (or explosion proof). They produce flow from 1,500 to 3,000 scfm and range from ½ hp to 1 hp.

  Estimated cost for enclosed motor -- $1,500 to $2,000 each

- First choice for more drive and large runs of ducting:

  TA 16 – tube axial blower with heavy-duty housing and "non-sparking" cast aluminum fan blades. They are also available with totally enclosed or explosion-proof motors. They flow 5,500 scfm of high drive air with a 2-hp electric motor.

  Estimated cost for enclosed motor -- $2,600 each.

- First choice for maximum cooling effect (larger area):

  Double-duty Heat Killer – axial vane, electric-driven blower with adjustable guide vanes to lower or increase the velocity and change the flow patterns. These are available with enclosed or explosion-proof motors. Designed for high performance cooling and effective on "air heat quenching." The recommended size would be the portable 24" K10 with a 1-hp electric motor moving 9,500 cfm.

These can also be equipped with an optional "Cold Front" evaporative cooler, "to lower the temperature of the cooling air." It is our opinion that this will not be needed.

Estimated cost with enclosed motor is $3,000 each.

For such applications as furnace cooling, you may find combinations of some of the above will be more effective. Certain units can be run as primary ventilators and later used for direct cooling.

☑ **RECOMMENDED PROJECT (#8)** – Remove all existing air horns from use and replace them with the appropriate number of electric units. Utilize the electric horns and train personnel in their use.

Equivalent number of air-operated air horns operating	10 units
Operating hours (summer season only)	2190 hours
Total operating cost: 10 air horns x 124 scfm each x 1 kW per scfm x 2190 hours in summer season x 5¢ per kWh	$27,156
Total operating cost of electric-operated air horns (negligible or < $1000)	$0
Equipment cost (10 units)	$20,000

4.5.2 Air-Operated Diaphragm Pumps

Although air-operated diaphragm pumps are not very energy efficient, they tolerate aggressive conditions relatively well and run without catastrophic damage even if the pump is dry. There are several questions to ask and areas to investigate that may yield significant air savings:

- Is an air-operated diaphragm pump the right answer? An electric pump is significantly more energy efficient. Electric motor driven diaphragm pumps are readily available. An electric motor drive progressive cavity pump may also well work.
- Consider installing electronic or ultrasonic controls to shut pumps off automatically when not needed. Remember that pumps waste the most air when they are pumping nothing.
- Is the pump running most of the time at the lowest possible pressure? The higher the pressure is, the more air is used. For example, filter-packing operations often do not need high pressure except during the final stages of the filter packing cycle. Controls can be arranged to generate lower pressures in the early stages and higher pressures later on, which may generate significant savings.

Air-Operated Diaphragm Pumps

Nominal Pump Size	Air Pressure Range	Nominal Scfm	Displacement (gals/cycle)	Estimated Electric Pump HP
1"	65 - 100 psig (average 80 psig)	25 – 30	0.08	¾ hp – 1 hp
< ½"	65 - 100 psig (average 20 psig)	15 – 20	0.09	¾ hp – 1 hp
1 ½"	70 – 100 psig (average 80 psig)	45 – 58	0.34	1 ½ hp – 2 ½ hp
2"	80 – 120 psig (average 95 psig)	90 – 120	0.43	3 hp – 5 hp
3"	90 – 100 psig (average 100 psig)	125 – 150	1.25	5 hp – 7 hp

- The above numbers are based on pumping water
- Flow varies by brand, model, and application
- Pressure requirement varies by brand, model, and application
- Air flow goes up as the flow and pumping cycles per minute increase
- Pressure requirement (air) may rise as head increases
- The electric pump horsepower will increase significantly at higher head.

All of the diaphragm pumps were smaller than 1 ½" and were run well controlled only as needed.

☑ **PHASE 2 RECOMMENDATION** – Review all air-operated diaphragm pump operating costs with those of an appropriate electric unit.

4.5.3 Air Motors and Hoists

Compressed air is a very inefficient transfer of energy requiring 7 to 8 hp of electrical energy to produce enough air to deliver 1 hp worth of work. For this reason, air hoists and air motors are often good targets to be replaced by electric-powered units. These applications use 15 to 20 cfm each or the equivalent of 5 hp worth of air.

Air hoists are rated in tonnage of capacity. Often the air motor horsepower is the same for several different tonnage ratings. Care should be taken to review the actual performance chart of the hoist in question.

☑ **PHASE 2 RECOMMENDATION** – Continue to look for air motor/hoist applications. Monitor incoming equipment.

4.5.4 Air Vibrators

Air vibrators are used to keep product or packaging moving or separated – e.g., keeping lids separated prior to sealing. If a plant employs air vibrators that use about 10 cfm each, they will require about 2.5 hp or more to produce the same as a similar electric vibrator, which might use about 0.25-hp input energy.

☑ **PHASE 2 RECOMMENDATION** – Continue to look for air air-operated vibrator applications. Monitor incoming equipment.